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INVESTIGATION OF THE USE OF MODERN SHIP MOTION PREDICTION MODELS IN IDENTIFYING SHIPS WITH A LARGER THAN ACCEPTABLE RISK OF DYNAMIC CAPSIZING

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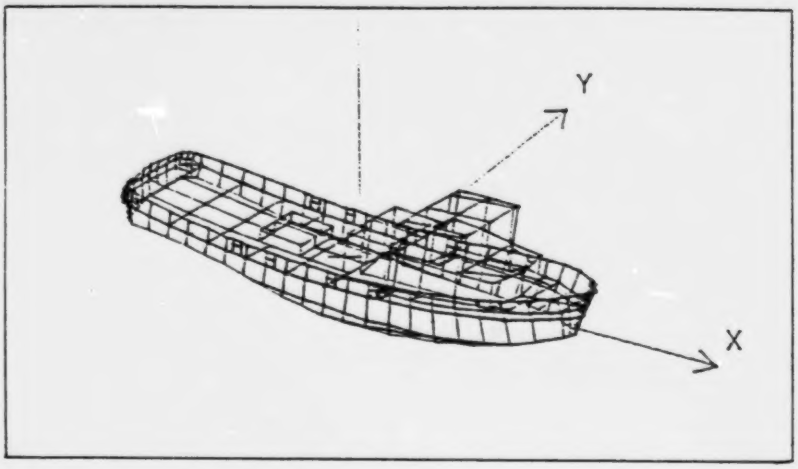
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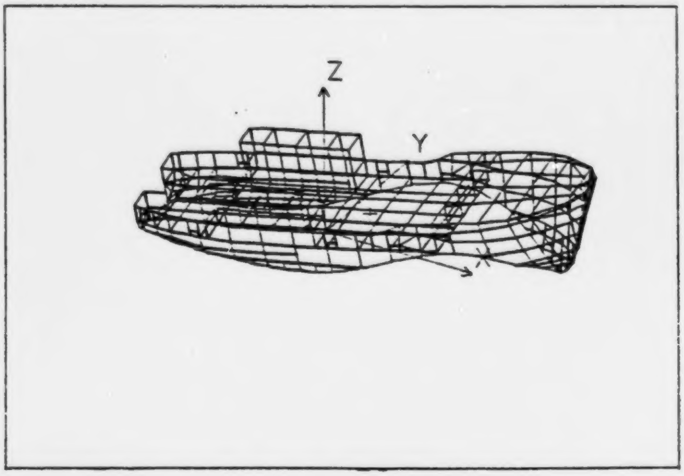
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17. Abstract

This report presents the results of the application of a suite of marine vehicle motion analysis computer programs to simulate the stability characteristics of fishing vessels. The programs utilized were developed by Atkins R&D (U.K.) for the use of the off-shore energy resources industry.

The performance of two 150-ton fishing vessels, one of a hard chine design and the other of a round bilge design, were simulated to compare the capsize behaviours of the two vessels in various sea conditions in an attempt to identify the influence of the different underwater hull forms.

The sea conditions and vessel speeds chosen for the computer simulations covered three known dangerous capsize mechanisms, vis: subharmonic rolling, resonant rolling and stability loss on a wave crest.

The results indicated that the round bilge vessel was somewhat more stable under the first two conditions, but both vessels capsized during prolonged presence in a wave crest while travelling at wave speed.

It was concluded that programs of the type used could be developed into useful tools to investigate the dynamic stability of vessels.

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17. Résumé

Ce rapport présente les résultats d'une série de simulations par ordinateur concernant les caractéristiques de stabilité des bateaux de pêche. Les programmes d'analyse utilisés avaient été élaborés par Atkins R & D (R.-U.) à l'intention de l'industrie des ressources énergétiques au large des côtes.

Les simulations ont porté sur deux bateaux de pêche de 150 tonneaux, différenciés par la forme de leur coque: coque à bouchains vifs dans un cas et coque à bouchains ronds dans l'autre. En comparant les caractéristiques de chavirement par différents états de mer, les chercheurs espéraient déterminer l'influence que pouvaient avoir différentes formes de carène.

Les états de mer et les vitesses d'évolution retenus matérialisaient trois conditions dangereuses connues pour être propices au chavirement, soit le roulis en mode sous-harmonique, le roulis en mode résonnant et la perte de stabilité sur crête de houle.

Les résultats montrent que le bateau à bouchains ronds est quelque peu plus stable que l'autre dans les deux premières conditions, mais que les deux chavireraient s'ils restaient assez longtemps sur une crête de houle et avançaient à la même vitesse que cette dernière. Le rapport conclut que les programmes du type utilisé pourraient, sous réserve d'une mise au point, servir utilement à l'étude de la stabilité dynamique des navires.

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SUMMARY

The aim of the study was to investigate whether the application of computer programs developed to simulate the motion of marine structures could prove useful in the analysis of fishing vessel stability characteristics.

The programs used are known as the AQWA suite of hydrodynamic programs developed by Atkins R & D of the United Kingdom for the use of the offshore energy resources industry.

Two 150 ton fishing vessels, of different hull forms, each similar to types which are operated in Canadian waters, were compared for their propersity to capsize. The first vessel was assumed to have a "hard chine" form and the second a "round bilge" type with fuller lines forward and aft.

The environmental conditions and vessel speeds chosen for computer simulations cover three known dangerous capsize mechanisms, viz : subharmonic rolling; resonant rolling; and stability loss on a wave crest. Selected computer runs were made for the two vessels when subjected to two wave heights and three wave directions.

On the basis of the limited simulations undertaken, it was concluded that :

1. Hydrodynamic programs such as those included in the AQWA suite utilized for this project, can be of value to regulatory agencies in investigating the dynamic stability of vessels in both the analysis of incidents and in the evaluation of proposed regulatory standards. Programs of this type can provide assistance in determining the hull form and stability characteristics to be avoided in order to reduce the risk of capsize. However, the size, complexity, availability and cost of such programs preclude the imposition of such programs in regulation.

2. The simulation undertaken assuming a subharmonic rolling in a wave encounter period of half the vessels natural roll period confirmed that because very regular waves of precise period are required, roll motions only grow to dangerous levels if unrealistically low levels of damping are assumed. The results also demonstrate that ensuring a large angle of vanishing stability is an effective measure to reduce the possibility of capsizing in this sea condition.

3. In the case of resonant rolling in a wave encounter equal to the vessels natural period it was shown that the "round bilge" vessel modelled has a greater dynamic stability in waves than the "hard chine" vessel which capsized at both levels of damping assumed, despite both vessels having the same GM. At angles below the angle of deck immersion (about 30 ° for both vessels investigated) the under-water hull form dominates the curves of the righting arms (GZ). This results from differences in hull designs.

Measures to reduce the danger of capsize would be to ensure a positive small angle tranverse stability in waves and to provide good damping characteristics (e.g. such as bilge keels). However the GZ curve for all angles of heel is of significance

4. The simulation of the vessels travelling at wave speed resulting in a prolonged presence on the wave crest showed that both vessels lost stability and capsized.

5. Safety of small vessels is strongly influenced by the attenuation of wave induced stability which the vessels experience when a wave crest is at amidships.

   The hull form aft plays a significant role on stability in certain positions relative to the wave profile. With the hull balanced on the crest, and with the stern emerging from the wave surface, the vessels lose waterplane inertia and hence stability

6. Increase in damping reduces the roll motions and can in turn produce greater dynamic stability and reduce the probablity of capsizing. However, in the capsize cases of the vessels riding on top of, and with the wave crest, damping has little effect.

It is recommended that within the framework of the present static stability rules, a modified regulation should be considered which would improve the safety of small fishing vessels by requiring a large angle of vanishing stability, i.e. positive GZ values up to 80 - 90 °, recognizing that small vessels will heel to much larger angles more often than large vessels.

To improve the safety of small fishing vessels considerations are also recommended to be given to :

(a) A requirement for a range of positive GZ at all vessel positions in waves.

(b) A minimum roll damping requirement.

In recognition of the limited data available from this and other studies, it may not be practical at present to formulate the regulations (a) and (b) above. Further case studies may be justified to expand the analysis with the objective of developing simplified assessment criteria for use in stability regulation. It is recommended that such studies be initiated to provide a technical basis to assist in the process of modifying regulations.

SOMMAIRE

Cette étude avait pour but de determiner si les programmes de simulation par ordinateur mis au point pour étudier le comportement des structures marines pourraient servir utilement à l'analyse des caractéristiques de stabilité des bateaux de pêche.

Les programmes visés sont connus par le sigle AQWA et constituent une série axée sur l'étude des phénomènes hydrodynamiques jouant sur les installations de forage en mer; ils ont été créés par Atkins R & D du Royaume-Uni.

La simulation a porté sur les caractéristiques de chavirement de deux bateaux de pêche semblables à ceux exploités couramment dans les eaux canadiennes; tours deux jaujeaient 150 tonneaux, mais se différenciaent par la forme de leur coque : coque à bouchains vifs, dans un cas, et coque à bouchains ronds, dans l'autre,cette dernière étant également caractérisée par une forme plus ronde à l'avant et à l'arrière.

Les états de mer et les vitesses d'évolution retenus matérialisaient trois conditions dangereuses connues pour être propices au chavirement, soit le roulis en mode sours-harmonique, le roulis en mode résonnant et la perte de stabilité sur crête de houle. Les diverses simulations réalisées ont pris en compte pour les deux bateaux deux hauteurs et trois directions de houle.

Voici les conclusions tirées des simulations limitées qui ont été faites :

1. Les programmes de simulation apparentés à ceux de la série AQWA utilisée pour fins de la présente étude peuvent aider les organismes de réglementation à déterminer les caractéristiques de stabilité dynamique des bateaux, aussi bien à l'occasion d'enquêtes sur incidents maritimes que pour l'évaluation de nouvelles règles de conception proposées. Ils peuvent aussi aider à définir les formes de coques el les caractéristiques de stabilité à éviter pour réduire les risques de chavirement. Toutefois, étant donné qu'ils sont peu accessibles, onéreux, complexes et volumineux, on ne saurait en imposer l'utilisation réglementaire.

2. En régime de roulis sours-harmonique correspondant à une période de rencontre de la houle égale à la moitié de la période propre de roulis des bateaux considérés, la simulation a indiqué que, comme la rencontre d'une houle très réguliere de période donnée était une condition essentielle pour qu'il y ait augmentation d'amplitude, celle-ci ne pouvait atteindre une valeur critique que si l'amortissement pris en compte était anornmalement faible. Cette simulation montre également qu'en donnant aux bateaux un angle de perte de stabilite important on se premunit efficacement contre la possibilité de chavirement dans les conditions considérées.

3. Dans le cas d'une période de rencontre de la houle égale à la période propre de roulis des bateaux considérés, la simulation a montré que celui à bouchains ronds offre une plus grande stabilité dynamique que le bateau à bouchains vifs, lequel a chaviré aux deux niveaux d'amortissement pris en compte, même si les deux bateaux présentent la même hauteur métacentrique. A des angles de gîte inférieurs à l'angle d'immersion du pont (environ 30 ° dans les deux cas), la courbe des bras de levier de redressement dépend principalement de la forme des carènes. On peut donc affirmer que les différences dans les caractéristiques de stabilité des deux bateaux s'expliquent par la forme différente de leur coque.

   Deux moyens de réduire le danger de chavirement seraient de réaliser un petit angle positif de stabilité tranversale en route libre et de prévoir une bonne caractéristique d'amortissement (à l'aide de quilles de roulis par exemple). Ceci dit, l'allure de la courbe des bras de levier pour l'ensemble des angles de gîte demeure un facteur important.

4. La simulation concernant l'avancement à une vitesse égale à la célérité de la houle, donc correspondant à une présence prolongée en crête de houle, aboutit dans les deux cas à une perte de stabilité suivie de chavirement.

5. La sécurité en mer des petits bateaux est largement tributaire de l'atténuation de l'instabilité due à l'action de la houle lorsque la crête de cette dernière coincide avec le milieu du bateau.

   La forme de la coque à l'arrière influence beaucoup la stabilité des bateaux dans certaines positions par rapport au profil de la houle. Lorsque les bateaux se trouvent au sommet de la houle et que leur arrière émerge, ils perdent le bénéfice de l'inertie de la surface de flottaison et deviennent par le fait même instables.

6. Un meilleur amortissement a pour effet de réduire l'amplitude des mouvements de roulis et, partant, d'augmenter la stabilité dynamique des bateaux, avec comme corollaire une probabilité de chavirement réduite. Par contre, l'amortissement a peu d'influence dans les cas où les bateaux avancent à la même vitesse que la houle et au sommet de celle-ci.

Il est recommandé d'envisager une modification des règlements actuels sur la stabilité des bateaux dans le sens d'une plus grande securité des petits bateaux de pêche en imposant un grand angle de perte de stabilité, donc, des bras de levier de redressement positifs jusqu'à 80 et même 90 ° de gîte, prenant ainsi en compte le fait que les petits bateaux ont tendance à connaitre des angles de gîte beaucoup plus importants que les gros navires.

Pour améliorer la sécurité des petits bateaux de pêche, il est également recommandé d'envisager ;

a) Un règlement imposant que les bras de levier de redressement se situent toujours dans une fourchette positive, peu importe la position du bateau par rapport à la houle,

b) un règlement imposant un amortissement minimal du roulis.

Compte tenu de la quantité limitée des données fournies par la présente étude et d'autres, une formulation définitive des règlements (a) et (b) préconisés ci-dessus serait peut-être prématurée. D'autres études de cas permettraient de pousser l'analyse plus loin et d'élaborer un jeu simplifié de critères d'évaluation aux fins de l'application de la réglementation sur la stabilité des bateaux. Il est recommandé d'entreprendre ces études afin de se donner les fondements techniques sur lesquels asseoir les modifications éventuelles de la dite réglementation.

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1 INTRODUCTION

1.1 Background

This report presents the results of a study by Arctec Canada Limited (ACL) under Supply and Services Canada (SSC) Contract 1SD84-00317 that was awarded in response to an unsolicited proposal by ACL (SSC-UP-A-411).

The aim of the study was to investigate whether the application of computer programs developed to simulate the motion of marine structures could prove useful in the analysis of fishing vessel stability characteristics.

The programs used are known as the AQWA suite of programs developed by Atkins R & D of the United Kingdom. These programs were developed for the use of the off-shore energy resources industry and have been purchased by many major oil companies in Europe, the United States of America and Canada. In the latter two countries the AQWA programs are sold and operated by the Arctec Incorporate, in which ACL is a member company, under an exclusive agreement with Atkins R & D . A summary description of the programs is given in the next section 1.2 .

Two 150 ton fishing vessels with differences in hull form, each similar to types which are in operation today in Canadian waters were selected for comparison of their propensity to capsize.

Sections 2 and 3 describe the methodology and procedures adopted. The results and analyses for each vessel are given in Sections 4, 5 and 6 for three sea conditions studies : subharmonic resonance encountering frequency; resonance encountering frequency; and with the vessels at the crest of a wave while travelling at wave speed respectively. Section 7 discussed the justification for the roll damping coefficients used in the simulations, while conclusions and recommendations are given in Sections 8 and 9 respectively.

Computer outputs of the simulations are contained in Volume 2.

1.2 Analysis Programs

The Atkins R & D's AQWA suite of hydrodynamic programs are described below :

AQWA-LINE	is a three dimensional diffraction/radiation program, capable of calculating the wave response of floating structures of arbitrary shape.
AQWA-FER	is a post-processor for AQWA-LINE, which tabulates significant response and load amplitudes in up to 20 user-specified wave spectra.
AQWA-LIBRIUM	is a program for finding the equilibrium position in six degrees of freedom, of one or more moored floating structures.
AQWA-DRIFT	calculates long-term time histories of vessel motions in irregular seas, including "slow drift motions".
AQWA-NAUT	calculates short-term time histories of vessel motions in regular waves, including non-linearities such as varying immersion of structures and tightening of mooring catenaries.
AQWA-PLANE	is a comprehensive graphics package for the whole AQWA suite, producing mesh visualization, time histories and frequency responses.

For the present study, programs AQWA-LINE and AQWA-NAUT were used, with AQWA-PLANE providing graphical output.

To begin the analysis, AQWA-LINE was used to perform a classical linear diffraction/radiation calculation of vessel response. The theory of this calculation is described in [1]. It is an exact calculation without empirical coefficients of any kind, but assumes that the wave height is very small and expresses the vessel motions as heave, pitch, roll etc. per unit wave height. Thus the motions in waves of any size are just linear extrapolations of the motions in very small waves, so that any sudden catastrophic event like capsize is ruled out.

Therefore the AWQA-LINE results (given for reference purposes in Appendix C in Volume 2) were not used in this study as such, but passed on to the nonlinear time-history simulation program AQWA-NAUT. This program takes the hydrodynamic data from AWQA-LINE and uses it in the following way:

1) The "added mass" calculated by AQWA-LINE (for the wave period in question) is lumped together with the vessel's mechanical mass, carried around with the vessel. However it changes its orientation, and is used to calculate inertial forces just as for a mechanical mass.

2) The "wave radiation damping" calculated by AQWA-LINE (for the wave period in question) is likewise used to calculate a damping force based on the vessel's velocity and angular velocity components in its instantaneous surge, sway, yaw etc. directions.

3) Similarly the "diffraction force" calculated by AQWA-LINE is applied in the vessel's instantaneous surge, sway, yaw etc. directions. Thus, for example the heave force is always applied parallel to the vessel's mast, whatever direction the mast is pointing in. As the vessel changes its heading (defined as the direction of the projection of the vessel's keel onto the horizontal plane), AQWA-NAUT interpolates between AQWA-LINE diffraction forces calculated for different wave headings. And, as the vessel moves forward through the waves, AQWA-NAUT continuously adjusts the phase of the diffraction force, so that for example if the vessel is travelling near to wave speed, the diffraction force will be given an appropriately long period corresponding to the long wave encounter period of the vessel.

The remaining force that the water exerts on the vessel is the force attributable to the undisturbed water pressure in the wave, which acts both to provide roll stiffness for the vessel, and a measure of wave excitation on it. AQWA-NAUT calculates this exactly, by finding the instantaneous wetted surface of the hull (assuming the wave profile is undisturbed) and integrating the undisturbed wave pressure over it. As well as displaying this total force, AQWA-NAUT also splits the force into two components - a "hydrostatic" component (see Appendix A for definition) due to the still-water pressure field (extended to be negative above the still-water line), and a "Froude-Krylov" component (see Appendix A for definition) due to the extra pressures caused by the wave action - and displays them separately. By contrast, the standard Naval Architectural convention [2] is to keep the two components together and calculate them on the approximation that the water pressure in an undisturbed wave increases with depth below the instantaneous water surface according to the same hydrostatic law that the pressure follows in still water. Allowance for the wave-induced modifications to this law is known as "Smith's Correction". AQWA-NAUT performs a "Smith's Correction" calculation automatically.

Given the above assumptions, AQWA-NAUT proceeds to find the motion of the vessel by integrating Newton's equation of motion for the trajectory of the vessel in three dimensions. It should be emphasized that all the above assumptions are to a greater or lesser extent approximations, and at various stages in the work two modifications to the data used by AQWA-NAUT were temporarily introduced:

1) Additional roll damping, beyond that calculated by AQWA-LINE, was added. This is because it is known that vessel roll is damped by a turbulent viscous drag mechanism in addition to the wave radiation mechanism modelled by AQWA-LINE.

2) The "diffraction force" calculated by AQWA-LINE was omitted. As described above, AQWA-NAUT adjusts the phase of this force to correspond with the wave encounter period, thus in particular the force will appear as a steady one when the vessel is travelling at wave speed (infinite encounter period). This is not physically reasonable because the diffraction force is caused by the pattern of diffracted waves produced by the vessel when it is stationary, hence there is no such pattern if the vessel moves at wave speed.

2. THE ANALYSIS VESSELS

2.1 Vessels

Two different designs of fishing vessels, which are in service today in the Canadian waters, were considered in this study. The first vessel of interest is a "hard chine" vessel, similar to one which capsized in the early seventies. It had fine lines, a sharp bow, and a large flat run and wide transom at the stern. The superstructure was positioned forward of amidships. This vessel had been subjected to a series of physical model tests in waves similar to the conditions that existed on the day of the capsizing.

The other vessel selected for the performance comparison assumed a "round bilge" type design with fuller lines forward and aft, and deep draft. The superstructure was located aft of amidships with a watertight forecastle above the main deck.

The principal particulars of these two vessels are given in Table 2.1.

As noted, the two vessels are not only different in hull form but also different in size, with the "round bilge" vessel being slightly longer and deeper than the "hard chine" one. Since our aim was to investigate if the difference in hull form has any effect on the dynamic stability of the vessels, it was decided to study the two vessels on the basis of the same principal dimensions and GM conditions. As the "hard chine" vessel is of the prime interest in this study, it was decided to scale the "round bilge" vessel to produce a "geosim" model of similar size and conditions to the "hard chine" vessel but with a hull form as that of the "round bilge" vessel.

Based on the body plan of the "round bilge" vessel supplied by the CCG, lines plans were constructed for the "geosim" model using the following scaling factors : 0.94 in ship length, 1.0 in beam, and 0.92 in depth. The resulting ship particulars of the "geosim - round bilge" vessel are also included in Table 2.1

TABLE 2.1 Principal Particulars of Analysis Vessels

Vessel	HARD CHINE VESSEL	ROUND BILGE VESSEL	GEOSIM-ROUND BILGE VESSEL
Length (OA) ,m	24.02	25.50	23.97
Length (BP) ,m	21.95	23.35	21.95
Length at WL ,m	22.40	23.78	22.40
Breadth (MLD) ,m	6.71	6.71	6.71
Depth (MLD) ,m	3.37	3.66	3.37
Draft Amidship ,m	2.32	2.52	2.32
Draft Forward ,m	1.85	1.86	1.74
Draft Aft ,m	2.78	3.18	2.92
Trim by Stern ,m	0.39	0.53	0.46
Rake of Keel ,m	0.53	0.79	0.72
Displacement ,tonnes	152.41	167.56	154.80
Transeverse GM ,m	0.52	0.55	0.52
Freeboard Bow ,m	3.63		3.57
Freeboard Aft ,m	1.87		2.10
Freeboard Mimimum ,m	1.76		1.95
Angle of Edge Immersion ,°	30.06		32.65
Waterplane coefficient	0.80		0.82
Block coefficient	0.44		0.44

2.2 AQWA Models

The AQWA programs require a discrete element model for the geometrical surface of each of the fishing vessels. For the "hard chine" fishing vessel, the model discretization was based on the line plans supplied by the CCG. Due to the vessel's geometric symmetry about its center line plane, only one half of the vessel was required to be modelled. The discretization of the wetted surface of the vessel requires a fine mesh model for the AQWA-LINE calculation. A coarse mesh model for the freeboard surface geometry of the vessel, including the bulwarks, freeing ports, main deck, and wheelhouse, was added on for the AQWA-NAUT time-history simulation. The complete AQWA model for the "hard chine" vessel consists of 246 diffracting elements and 427 Morision points (or non-diffracting elements). The discrete element mesh is given in Figures 2.1 and 2.2 at different viewing points. Figure 2.3 shows the bottom flat run at the stern.

Based on the scaled-down lines plan, and using the same modelling criteria, an AQWA model was generated for the "geosim round bilge" fishing vessel. The forecastle on this particular vessel, as well as the other appendages such as the bulwarks, freeing ports, main deck and wheelhouse were also included in the model. The complete model, in this case, consists of 324 diffracting elements and 379 Morsion points. The discrete element mesh is shown in Figures 2.4 and 2.5 at different viewing positions. Figure 2.6 shows the general underwater hull form.

The fixed reference axes (FRA) are a conventional right-hand Cartesian coordinate system located at a position vertically in alignment with the vessel's center of gravity in the still water surface with the positive Z-axis pointing vertically upward and the positive X-axis pointing forward along the vessel's initial position. The local structure axes (LSA), which are the analysis axes system, are always parallel to the FRA, with the origin locating at and moving with the center of the gravity of the vessel. According to this convention a following sea is advancing along the positive x-axis, thus wave direction=0 ° ; and a beam sea is advancing along the positive y-axis, thus wave direction=90 ° (see Figure 2.7).

Translational motions of the vessel in the X, Y, and Z directions are surge, sway and heave and are positive in the positive direction of their respective associated axes. The rotational freedoms about X, about Y, and about Z are roll, pitch and yaw and are positive in a clockwise direction rotating along the coordinate axes.

The SI unit system was used in the program calculation: force in Newtons, moment in Newton-metres, length in metres, mass in kilograms, time in seconds.

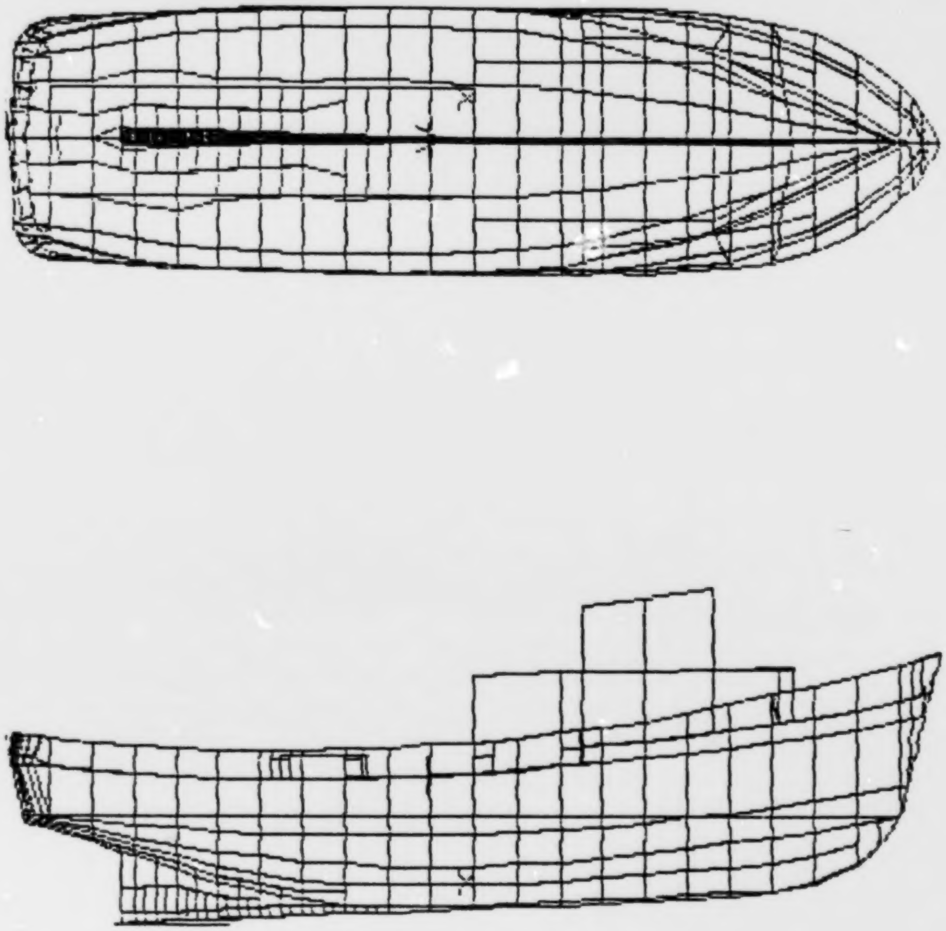


Figure 2.1 Projected Views of "Hard Chine" Vessel

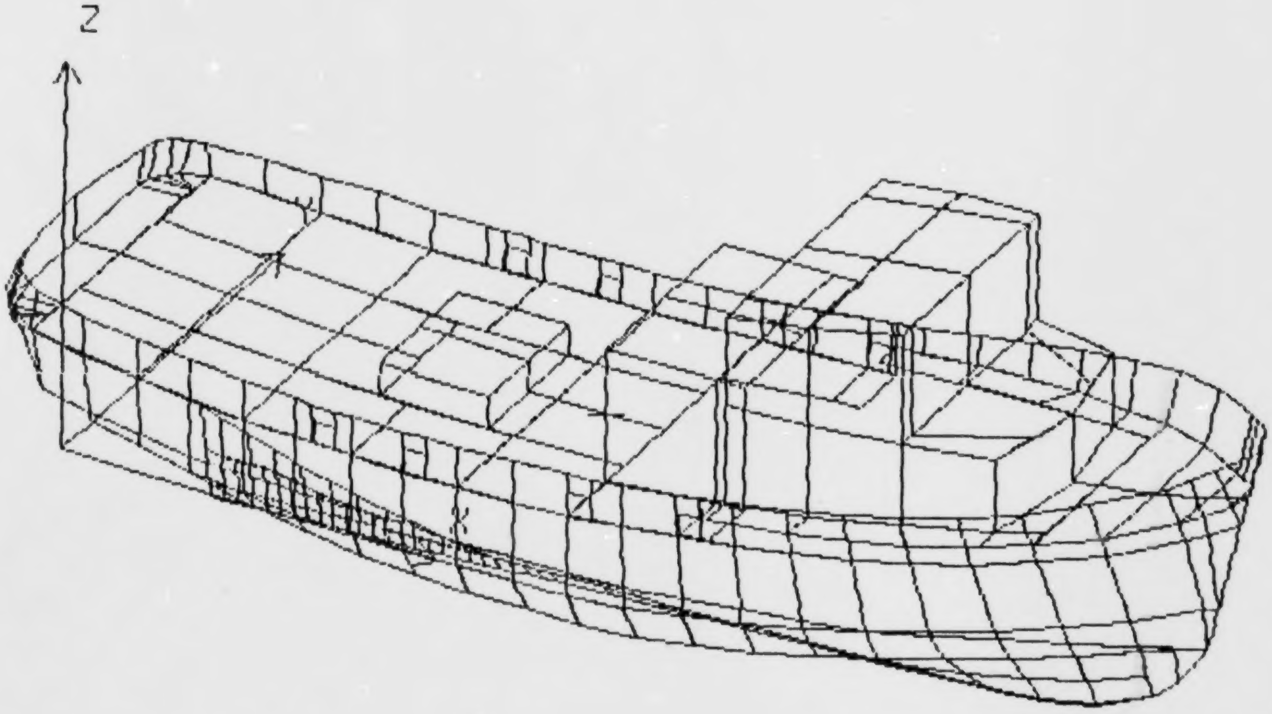


Figure 2.2 Isometric View of "Hard Chine" vessel

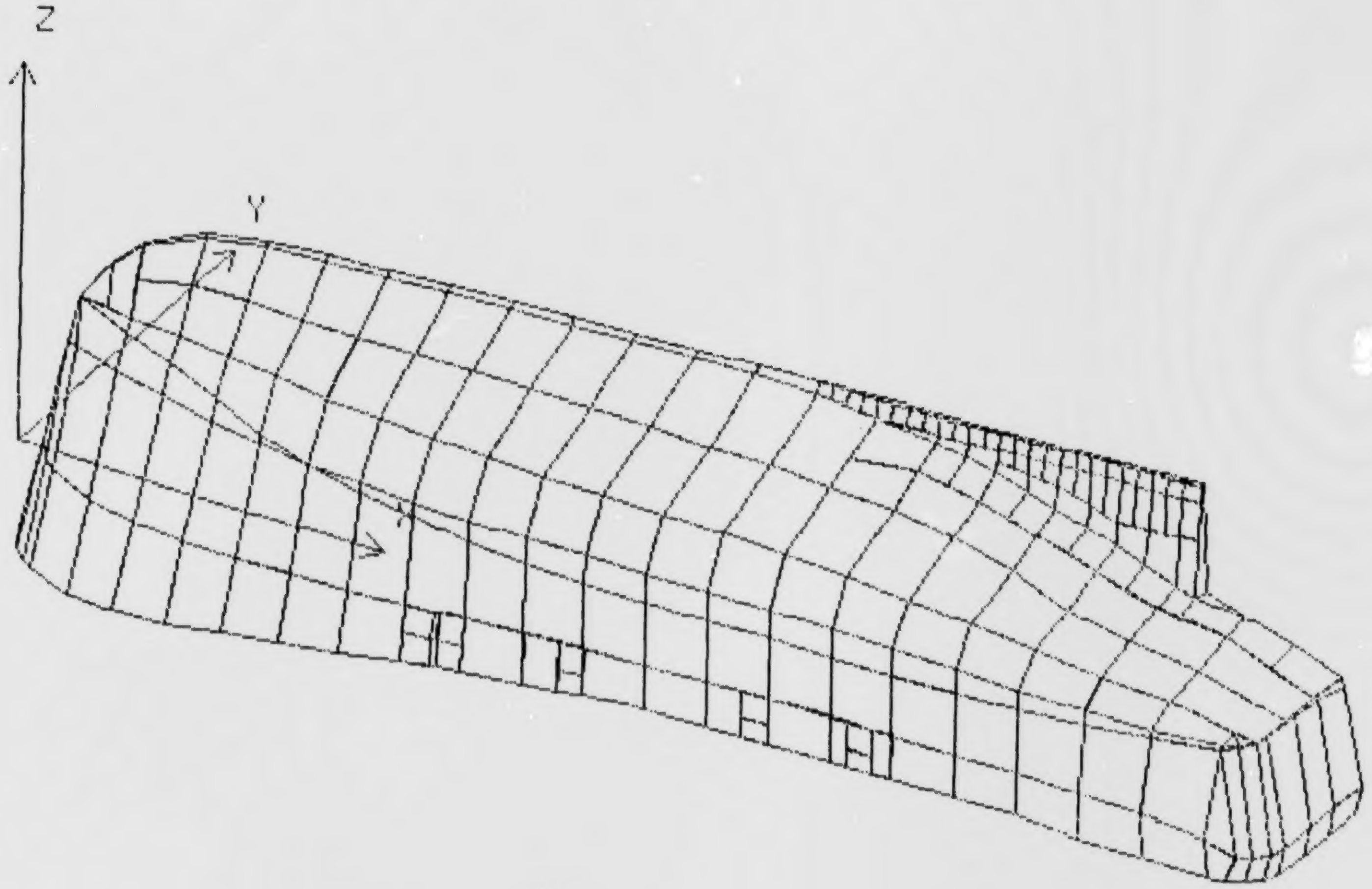


Figure 2.3 Stern View of Bottom Flat Run of "Hard Chine" Vessel

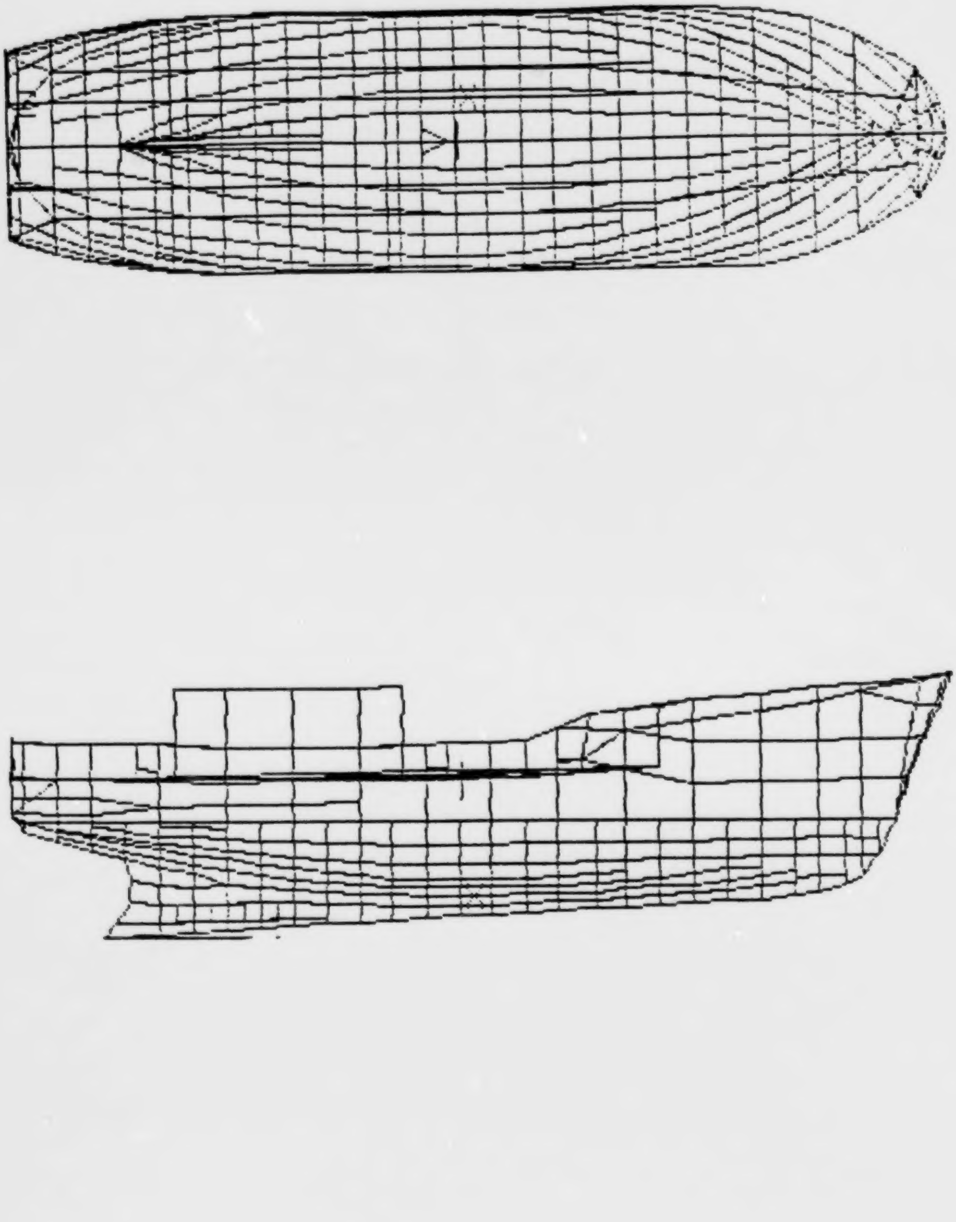


Figure 2.4 Projected Views of "Round Bilge" Vessel

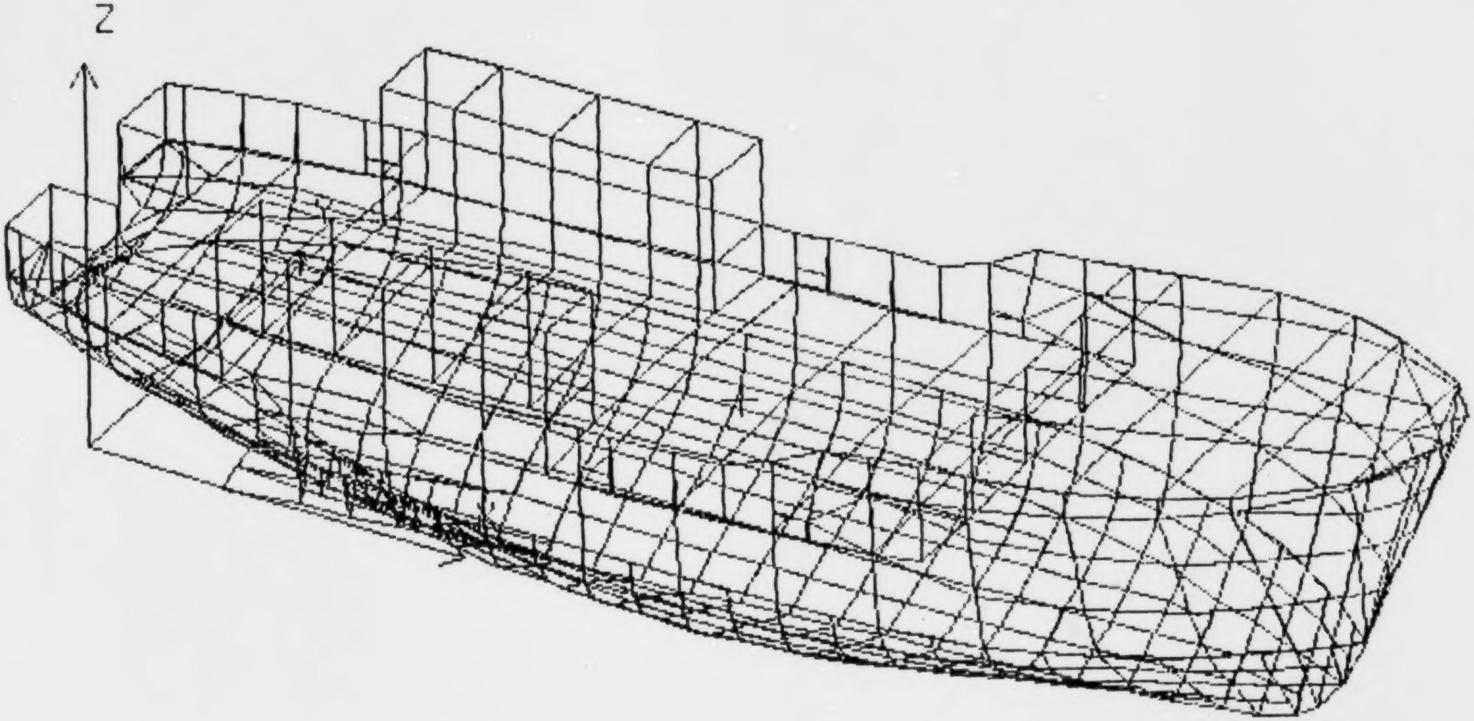


Figure 2.5 Isometric View of "Round Bilge" Vessel

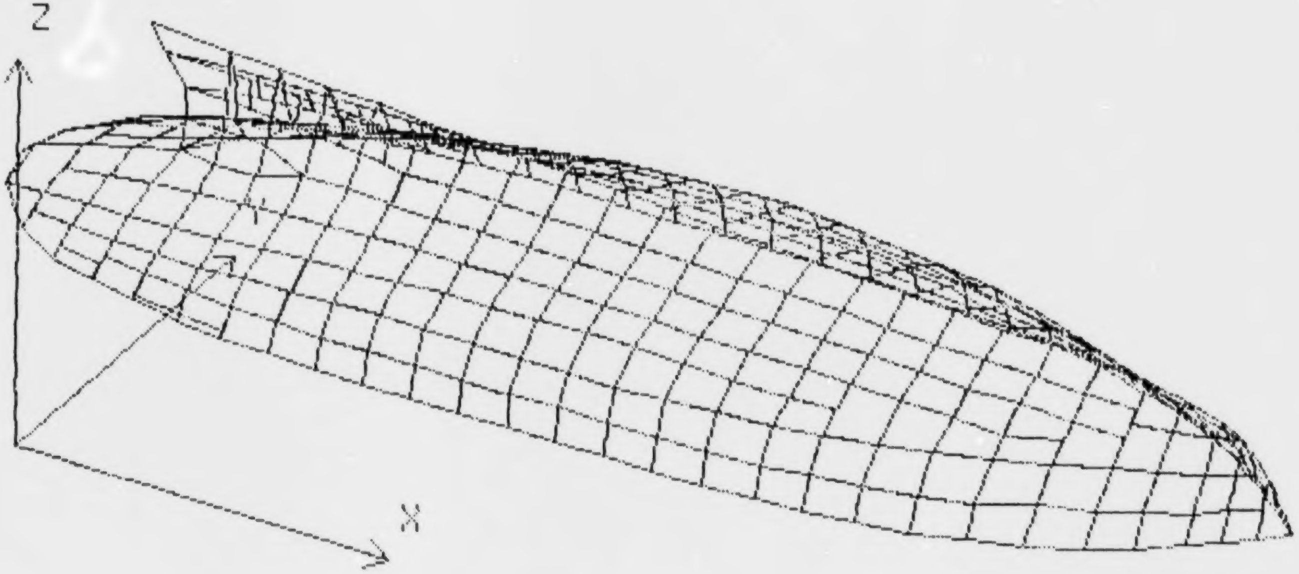


Figure 2.6 Isometric View of Underwater Hull Form of "Round Bilge" Vessel

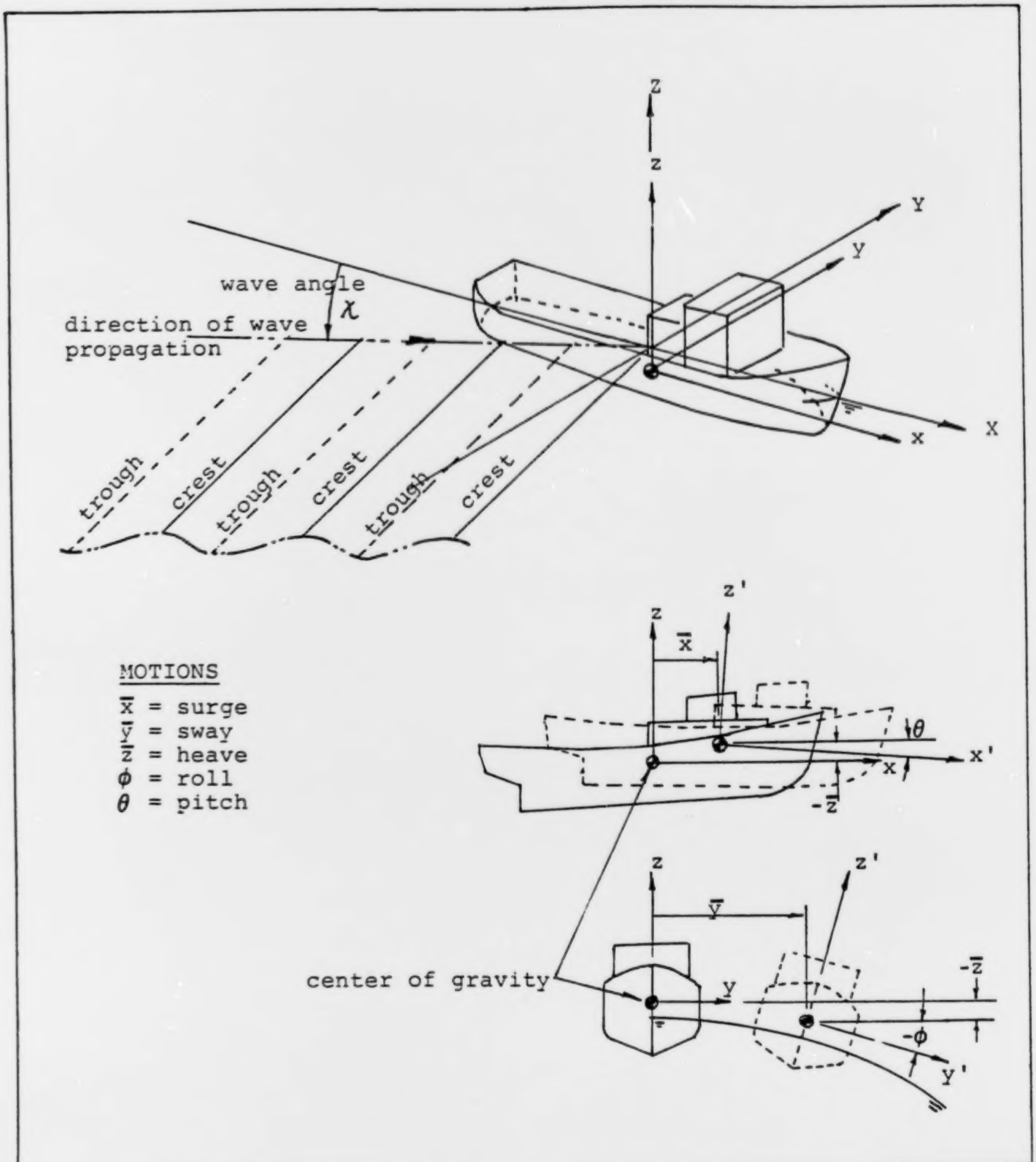

Z
z
Y
y
wave angle
χ
direction of wave
propagation
trough
crest
trough
crest
trough
crest
x
X
MOTIONS
$\bar{x}$ = surge
$\bar{y}$ = sway
$\bar{z}$ = heave
φ = roll
θ = pitch
z
z'
$\bar{x}$
θ
x'
x
$-\bar{z}$
z
z'
$\bar{y}$
center of gravity
y
$-\bar{z}$
-φ
y'

Figure 2.7 Definition Sketch for Coordinate System, Wave Directions and Motions

2.3 Stability Characteristics

The righting arms (GZ curves) in calm water for both the "hard chine" and the "round bilge" vessels were determined using the program AQWA-NAUT. The still water GZ curves calculated are given in Figure 2.8.

The initial stability for both vessels is essentially the same as each vessel has the same GM. For the "hard chine" vessel, the righting arm peaks at about 20 ° and vanishes at 38 ° . In striking contrast, the righting arm is positive for the "round bilge" vessel case to angles much greater than 60 ° , angles at which the "hard chine" vessel has a large negative righting arm. The large range of stability noted for the "round bilge" vessel is probably attributable to its watertight forecastle at the main deck.

The effect of waves on transverse static stability has been studied in [2,3] . In general the righting arms (GZ) are reduced below the still water values when the crest of a wave is near amidships, and increase above the still water values when a trough is near amidships.

In order to estimate the variations of stability in waves for the subject vessels, calculations were made assuming a wave period of 3.423 second and a wave height of 1.22 metres. GZ curves were calculated both for the wave crest and trough at amidships positions. The GZ curves obtained for each wave positions together with the curve for the still water condition are given in Figures 2.9 and 2.10. The curves in these figures do indeed support the above noted phenomena.

Figure 2.11 compares the stability in wave for the two reference vessels. With wave crest amidships, the righting arm of the "hard chine" vessel vanishes at 32 ° , but the range of stability of the "round bilge" vessel extends up to the large angle of 60 °. The comparison indicates that the "round bilge" vessel produce greater transverse stability at large angles of roll than the "hard chine" one.

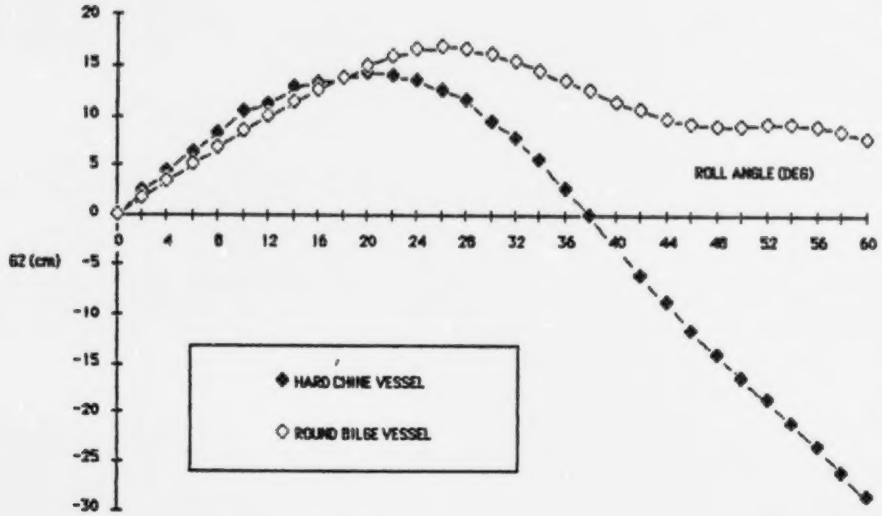


Figure 2.8 Still Water GZ Curves

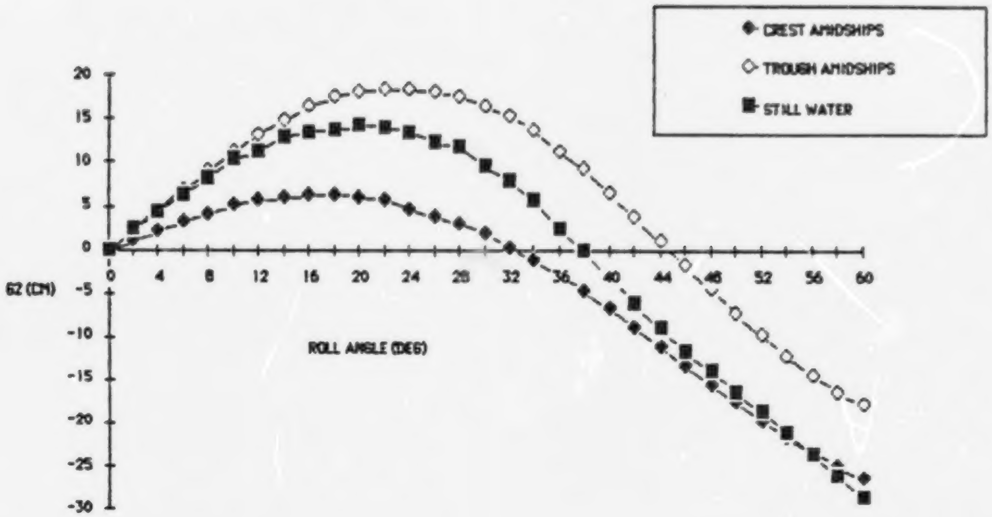


Figure 2.9 GZ Curves in Waves for "Hard Chine" Vessel

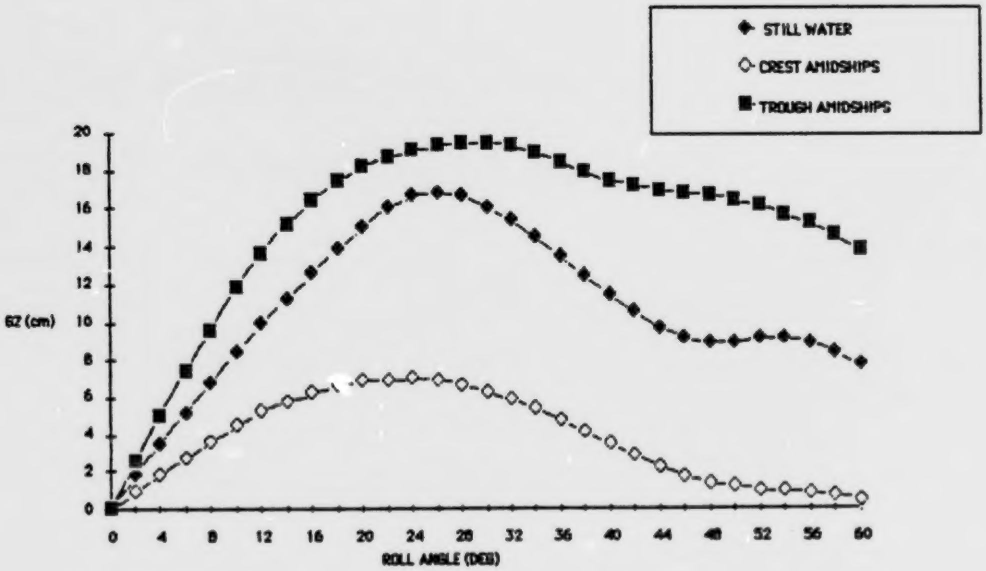


Figure 2.10 GZ Curves in Waves for "Round Bilge" Vessel

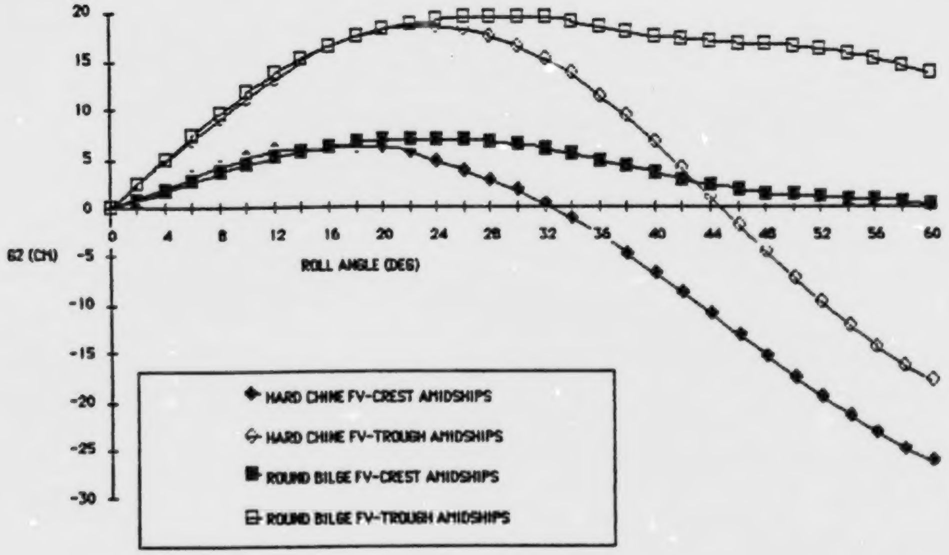


Figure 2.11 Comparison of GZ Curves in Waves

3. SCHEME OF STUDY

3.1 Rationale

With the geometry and weight distributions of the two vessels fixed as described in Section 2, considerable attention was next paid to devising a rational scheme for investigating capsizes. Videotapes of physical model tests carried out by NRC on models of similar vessels were examined, which, taken together with much more extensive model test evidence found in the literatures [2,3,4], led to the conclusion that three wave induced capsize mechanisms should be the focus of the study :

1) Subharmonic resonance or "Mathieu instability" when the wave encounter period is precisely one half of the vessel's natural roll period. This mode of capsize was particularly noted in the NRC videotape, and was simulated with the program AQWA-NAUT as an initial test case.

2) Resonance in roll caused by a wave encounter period close to the vessel's natural roll period. This mode of capsize is noted in [2,4] .

3) Loss of stability due to prolonged presence in the crest of a wave, occuring when the vessel is travelling at or near the speed of the waves so as to have a very long wave encounter period. This mode of capsize is also noted in [3,4] .

It was accordingly decided with the Client to choose three sets of conditions for the AQWA-NAUT simulations which homed in on these three mechanisms. Since all the evidence suggested that waves of length close to the vessel length posed the greatest danger, only this wavelength was chosen, with two different wave heights, and three different headings. The vessel speed was then chosen so as to achieve the wave encounter periods of (1), (2) and (3) above. Finally, to limit expenses, the "hard chine" vessel was investigated first, in all these 18 conditions, and the "round bilge" vessel was, with some exceptions where indicated, subjected only to the conditions which had proved the most troublesome. The details of the conditions chosen are given in 3.2 below.

The initial test runs mentioned above revealed an important practical problem with the computer simulations, which was that of holding the vessel at the required heading and speed - exactly the same problem as is encountered during physical model tests, when considerable concentration is needed to keep a radio-controlled model running at the required speed without broaching.

After some unsuccessful experimentation using wind forces to hold the vessel (effectively giving it large "sails"), it was decided to implement an explicit control system in the AQWA-NAUT simulations, which was achieved by arranging for the program's axis system to travel along at the desired speed and direction of the vessel, and then add stiffness and damping elements in the surge, sway, and yaw freedoms of the vessel, so as to hold it close to zero position in these axes. These stiffness and damping elements were chosen initially to give the vessel a well-damped natural period of 20 s in those freedoms, corresponding to the helmsman achieving a control response of around 10 s . It was established that the forces placed on the vessel by these additional elements (which are lumped by the program in with the "hydrostatic" forces in the case of the stiffness elements, and with the "linear damping" forces in the case of the damping elements) were within the actual range of the engine power and rudder loading.

This control system (henceforth referred to as "Servo Design 1") proved adequate for case (1) above; for cases (2) and (3) it proved necessary to shorten the natural periods to 10 seconds to prevent the vessel from broaching in the largest waves (henceforth as referred to "Servo Design 2"). The associated heavy control forces, which in some cases were on the limit of what could in practice be achieved on the vessel, are a reflection of the difficulties experienced in practice at sea in preventing broach under these conditions.

As a check on the effectiveness of the servo designs against broaching, the corrective moments applied in yaw for both vessels in the steep (1/7L), following 30 ° wave condition were calculated and compared with the excitation moments. Details are given in Appendix B. The close comparison indicates that the control systems were perfectly satisfactory.

3.2 Analysis Parameters

The global environmental parameters that are held constant for all program calculations are given as follows:

- standard gravitation acceleration = 9.81 m/s
- water density = 1,025 Kg/cubic metre

For the water depth of interest, the waves are considered deep water waves and therefore the second order deep water theory was used in the dynamic wave pressure calculations.

The input parameters for the basic diffraction/radiation calculations of vessel motions in regular harmonic wave environments (AQWA-LINE) are essentially the wave periods and directions. In our case, they are as follows:

- six wave directions - 0 ° (stern sea), 30 °, 50 °, 90 ° (beam sea), 135 °, and 180 ° (head sea).
- four wave periods - 5.36, 4.19, 3.83, and 3.42 seconds.

The upper limit of the wave periods was chosen to give a wave length twice the ship length, a wave condition corresponding to the NRC video tapes where the sub-harmonic roll motions were observed in the model tests. The lower limit of 3.42 s represents the wave in the NRC tests where the physical model capsized.

As for the time-history simulation, the wave conditions were chosen in consultation with the Client as described in Section 3.1 above. The base case is a regular wave of 3.83 s period, giving a wave length of 1.05 times the ship length. Two wave heights have been considered : 1.910 and 3.24 metres, corresponding to wave height to length ratios of 1:12 and 1:7; and the wave directions are 5 °, 30 °, and 50 °. The weather is on the starboard side of the vessel.

Wind and current were not considered in the time history simulations.

Additional input parameters are the ship speeds. They were carefully chosen to give the specific encountering periods in the waves. The encountering periods are 0.5 and 1.0 times the vessel's natural roll period (Tr), and 100 s (nearly infinite). The aim was to simulate the three capsizing mechanisms noted in Section 3.1 above.

4. DYNAMIC BEHAVIOUR AT SUBHARMONIC RESONANCE ENCOUNTERING FREQUENCY

4.1 General

For these runs the speed was adjusted to produce an encounter period equal to half the natural period as calculated in AQWA-LINE. All runs were performed using "Servo Design 1". Since this capsize mechanism is inhibited by damping (see Section 4.2 below), all runs were initially performed with roll damping as calculated by AQWA-LINE which is the damping from wave radiation only and is just 0.4% critical *. The run (WM4) which produced a capsize was then repeated with more realistic (5% critical) level of roll damping and did not capsize.

HARD CHINE VESSEL

Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
WM1	5.0	1/12	0.4
WM2	30.0	1/12	0.4
WM3	50.0	1/12	0.4
WM4	5.0	1/7	0.4
WM5	30.0	1/7	0.4
WM6	50.0	1/7	0.4
WM4-5%	5.0	1/7	5.0

ROUND BILGE VESSEL

Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
EM4	5.0	1/7	0.4
EM4-5%	5.0	1/7	5.0

---

* The value of damping of a structure is often expressed in terms of a percentage of the critical. The value of the damping coefficient when the damping is critical is $C_c = 2\,(MK)^{0.5}$ where M is the mass and K is the stiffness. Stiffness of a structure originates basically from its material properties, and in general term, it is a load/deformation characteristics of the structure. For a ship the roll stiffness is given as $\rho g \nabla GM_T$, where ρg is the water weight density, ∇ is the volumetric displacement and GM_T is the transverse metacentre height.

4.2 Description Of Capsize Mechanism

The mechanism of this mode of failure is known as the First Mathieu Instability. This is an oscillatory rolling motion that builds up over a few cycles leading to capsize in roll. The theoretical requirements [5] for this to occur are strong fluctuations in the vessel's roll stiffness caused by regular waves with an encounter period half the vessel's natural roll period, and a low level of roll damping.

The mechanism can best be explained in energy terms. As the vessel rolls away from the vertical, it is doing work against the roll stiffness. As it rolls back to the vertical, the roll stiffness does work on it.

With a normal constant-stiffness roll "spring" the two are equal- i.e. the energy stored in the spring as it compresses is returned without loss as it expands. However, if there is a reduction in roll stiffness whenever the vessel is rolling away from the vertical, and an increase in roll stiffness whenever it is rolling back (which means two cycles of stiffness variation per cycle of roll motion), the spring puts more energy into the vessel than it takes out, so, provided this net energy input is greater than the energy losses through damping, the roll motions grow.

4.3 Results of Subharmonic Case

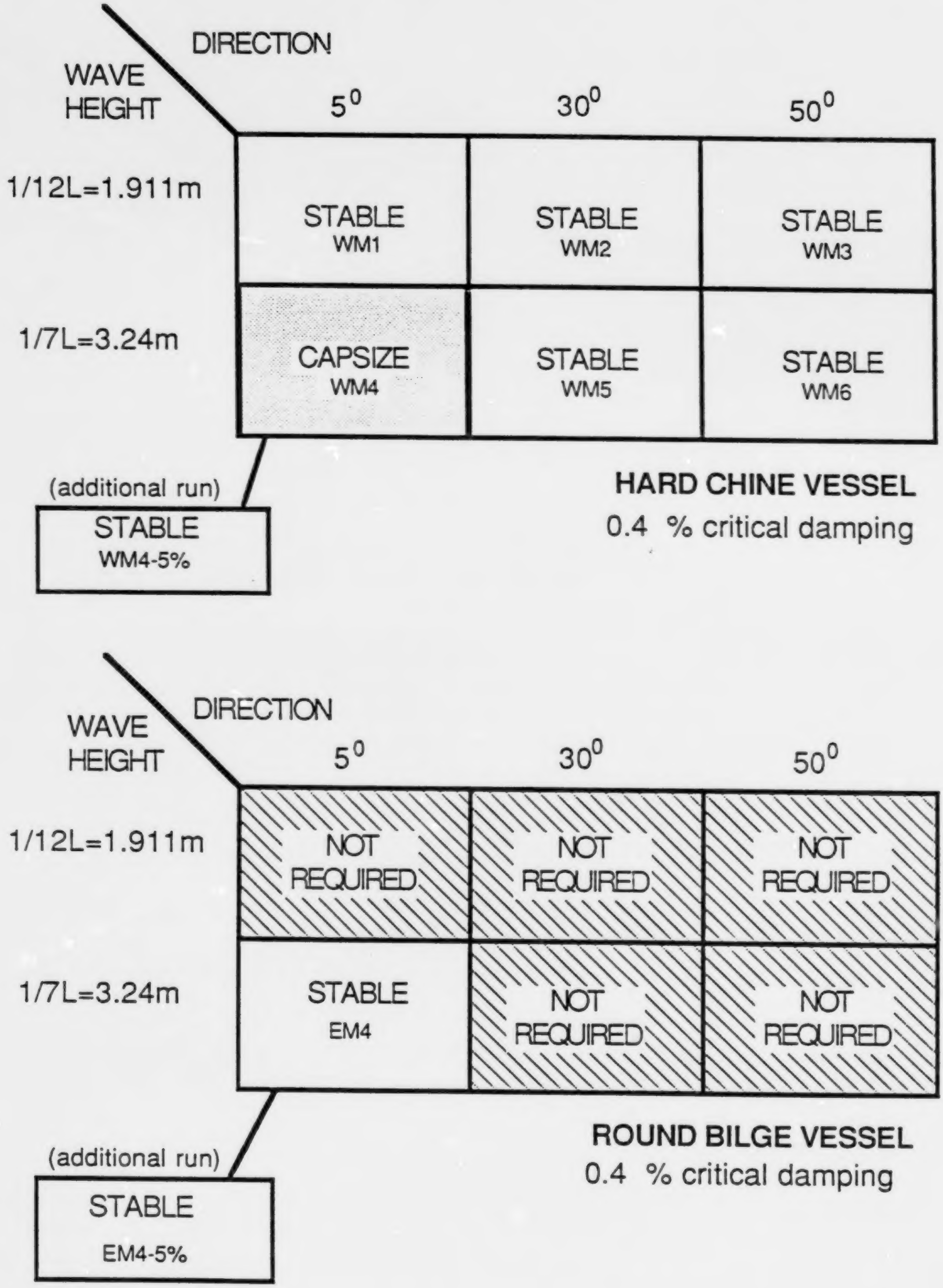
A summary of the results is given in Figure 4.1 and a complete set of the time motion histories for all runs are included in Appendix D.

With a 0.4 % critical damping in roll, the "hard chine" vessel survived all wave conditions, except the case WM4 in which the vessel capsized in a 3.24 m high, 5 ° stern sea. With the roll damping increased to 5 % critical, the vessel became stable.

As for the "round bilge" vessel, it survived the sea condition which appeared to be troublesome for the "hard chine" vessel, both with 0.4 % and 5 % critical roll damping.

4.4 Discussion of Results

The complete motion time history, in all 6 degrees of freedom for the capsize case WM4 is shown in Figure 4.2 . Figure 4.3 shows the time history in heave (z direction) and in roll (about x) from the start to 78 s as the vessel is rolled over to 50 ° on its way to capsize. From this plot it can be seen how the roll motion rapidly builds up from zero to 30 ° in 6.5 wave cycles (28 s). The roll then is reduced as the vessel's phase relationship with the wave changes due to "nonlinear detuning", i.e. due to its natural period lengthening for large roll angles



WAVE HEIGHT \ DIRECTION	5^0	30^0	50^0
1/12L=1.911m	STABLE WM1	STABLE WM2	STABLE WM3
1/7L=3.24m	CAPSIZE WM4	STABLE WM5	STABLE WM6

(additional run)

STABLE
WM4-5%

HARD CHINE VESSEL

0.4 % critical damping

WAVE HEIGHT \ DIRECTION	5^0	30^0	50^0
1/12L=1.911m	NOT REQUIRED	NOT REQUIRED	NOT REQUIRED
1/7L=3.24m	STABLE EM4	NOT REQUIRED	NOT REQUIRED

(additional run)

STABLE
EM4-5%

ROUND BILGE VESSEL

0.4 % critical damping

Figure 4.1 Results Summary for Mathieu Instability Case

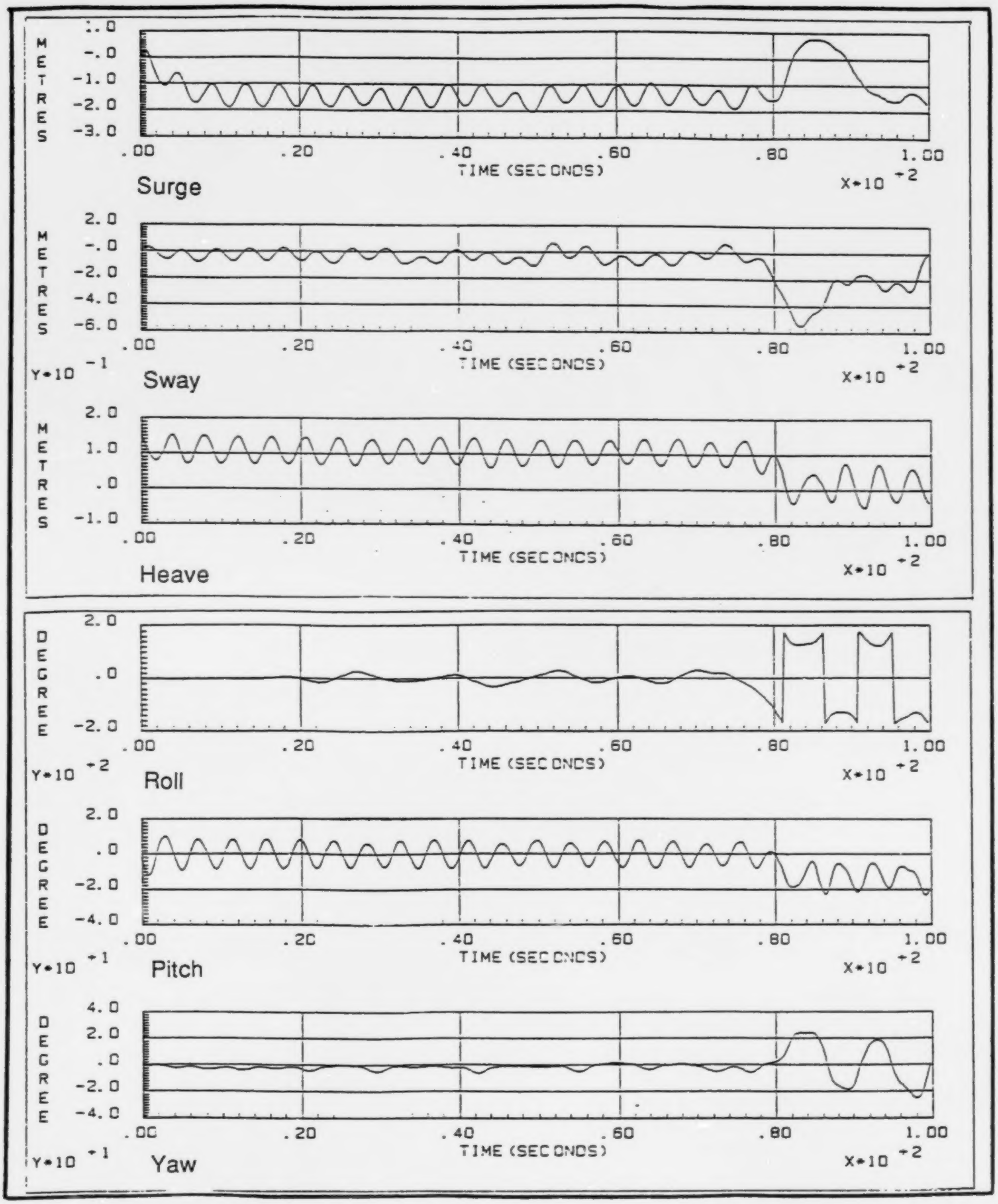


Figure 4.2 Motion History of "Hard Chine" Vessel in Case WM4

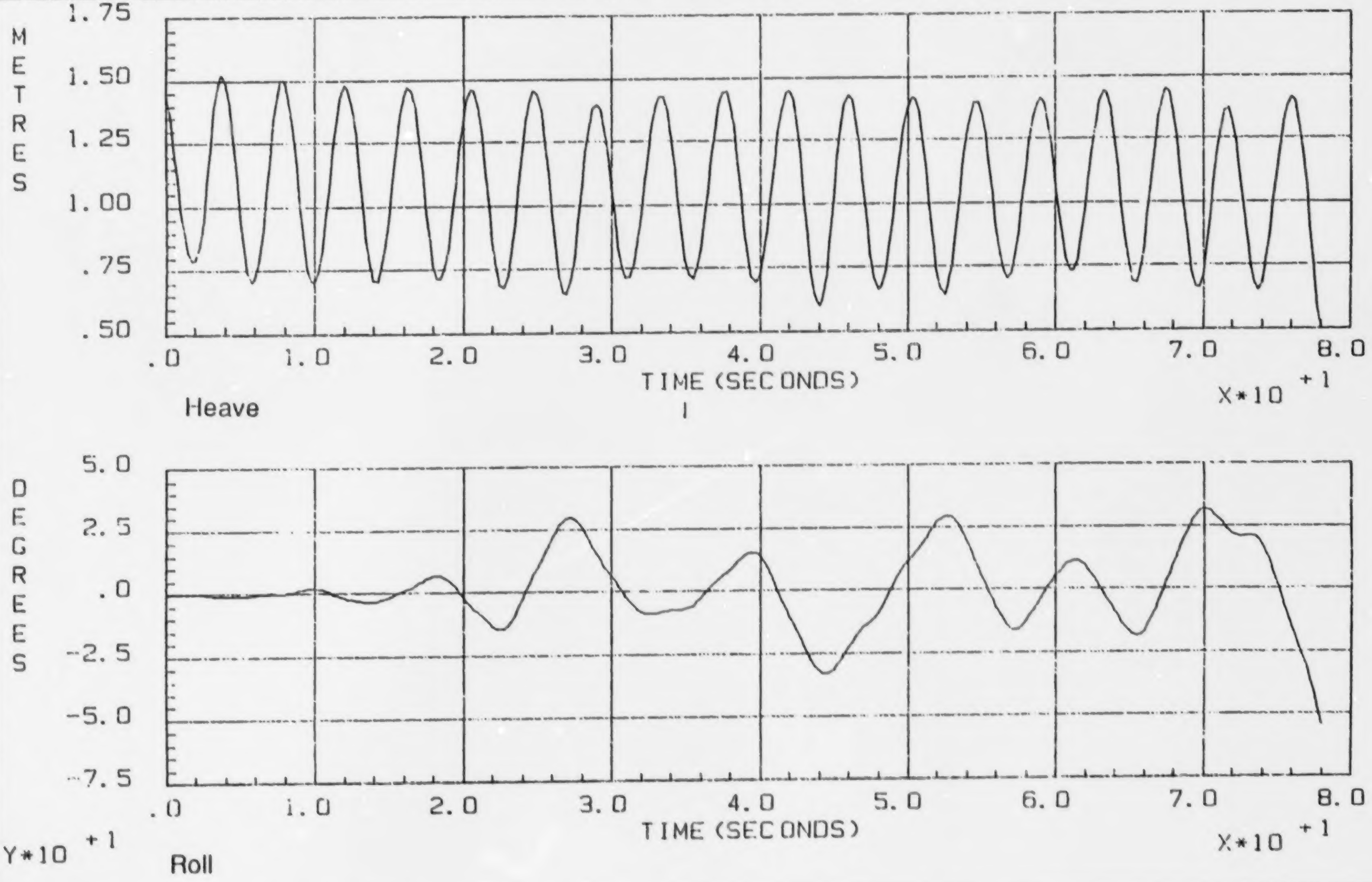


Figure 4.3 Heave and Roll Motions of "Hard Chine" Vessel for Case WM4

because of the loss of righting lever. Then from 40 s onward the roll motion builds up again leading to capsize.

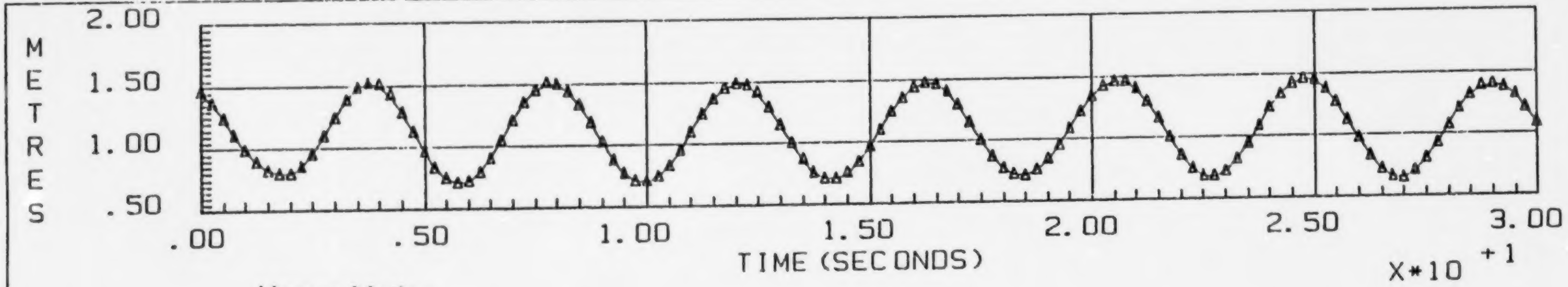
Figure 4.4 is an enlarged view of the heave and roll motions plus the roll velocity for the time period from zero to 30 s . It can be clearly seen that the roll amplitude is rapidly increasing. This figure also shows the phase relationship between the heave position and roll motion. At 14 s, the vessel is in a trough and starting to recover from a port roll angle (velocity=0) and heeling back to equilibrium. As the vessel reaches and passes through heave and roll equilibrium at 15.5 s . (reduced stiffness) the velocity reaches a maximum. Since the vessel was pulled back to equilibrium with a stiffer "spring" (14-15.5 s) than the spring resisting the roll away (15.5-18 s) the roll amplitude increases. Between 17 s and 19 s the vessel enters a trough, the roll velocity becomes zero, and the roll angle reaches a peak.

Capsize occurs when the vessel is heeled over to starboard far enough, that the restoring force at the trough is not able to right the vessel but simply brings it to rest briefly in a heeled over position as the next crest passes. When the vessel enters the next trough the restoring force accelerates it over with such force that as it rolls through equilibrium and over to port the restoring force is unable to prevent the vessel from rolling over and capsize occurs (Figure 4.3). Figure 4.5 shows the sequence of the capsize, every 20 s from the start and then every 1 s from 74 to 80 s.

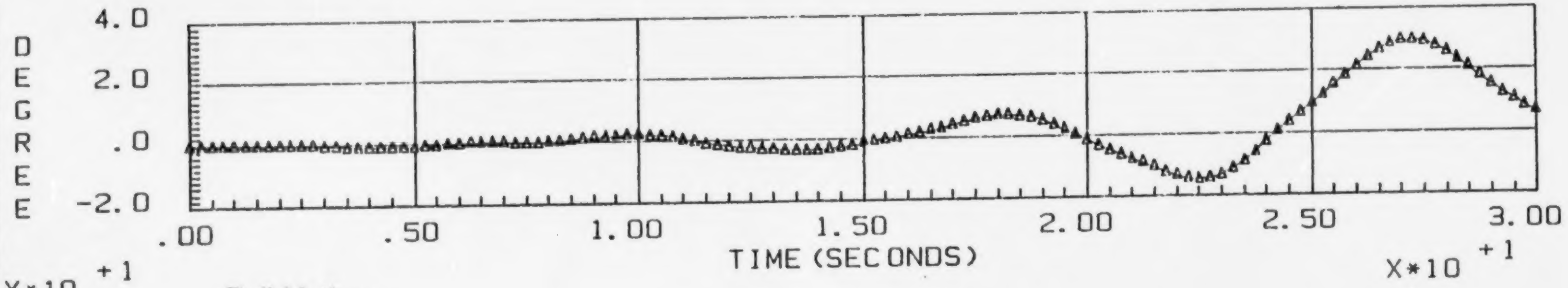
Figures 4.6 shows the results when the identical run is repeated with roll damping increased to a level of 5% critical. It can be seen that a capsize no longer occurs.

With the "round bilge" vessel case EM4, there is no capsize even without the additional damping. As can be seen from Figure 4.7, the roll motion grows to a large angle of about 55 ° at the end of the simulation (at 132 s) and yet the vessel recovers on the next wave and begins another buildup of roll motions.

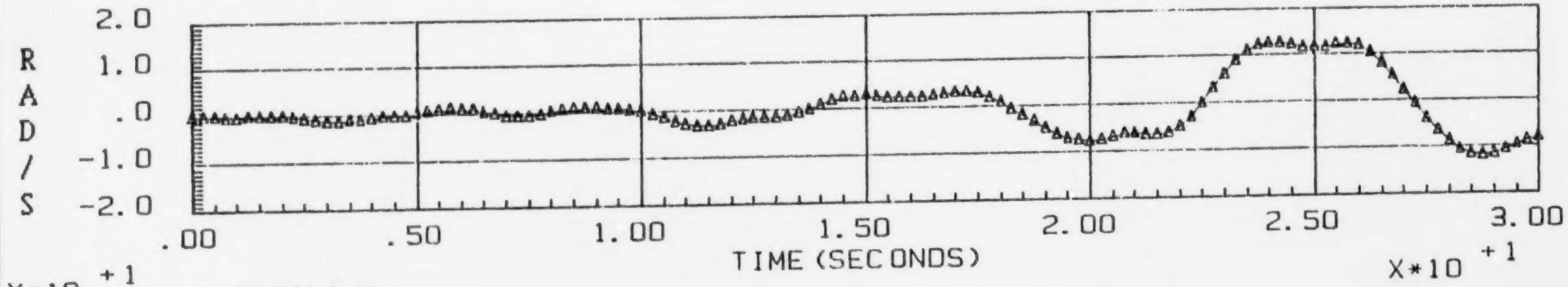
The fact that the "round bilge" vessel survived the steep wave, while the "hard chine" vessel did not, is presumably attributable to the superior righting-arm characteristics of heel of the "round bilge" vessel at large angles .



Heave Motion



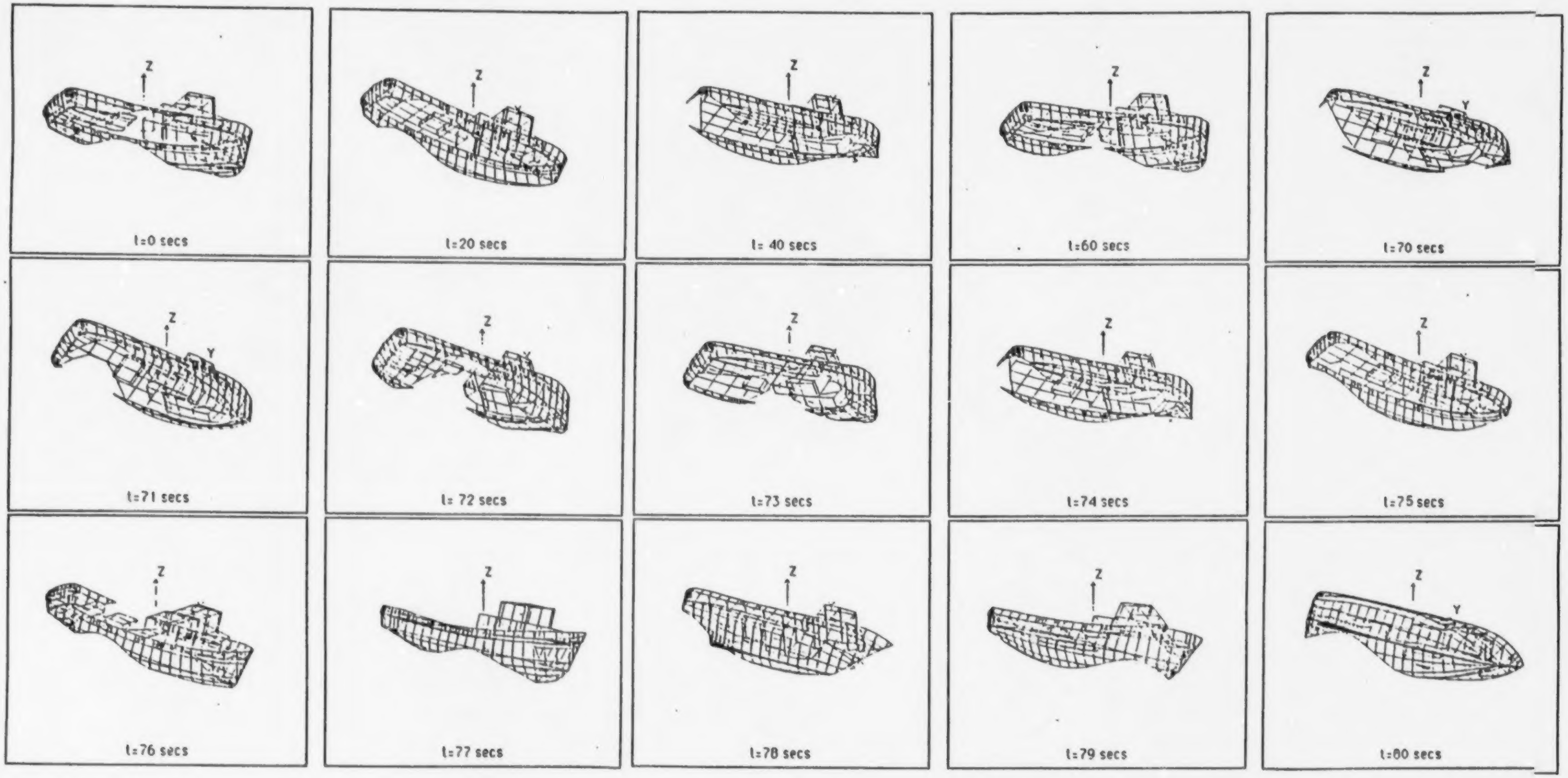
Roll Motion



Roll Velocity

Figure 4.4 Initial Transient Heave and Roll Motions, and Roll Velocity for Case WM4

Figure 4.5 Capsize Sequence of "Hard Chine" Vessel in Case WM4


Z
Y
t=0 secs
t=20 secs
t= 40 secs
t=60 secs
t=70 secs
t=71 secs
t= 72 secs
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t=79 secs
t=80 secs

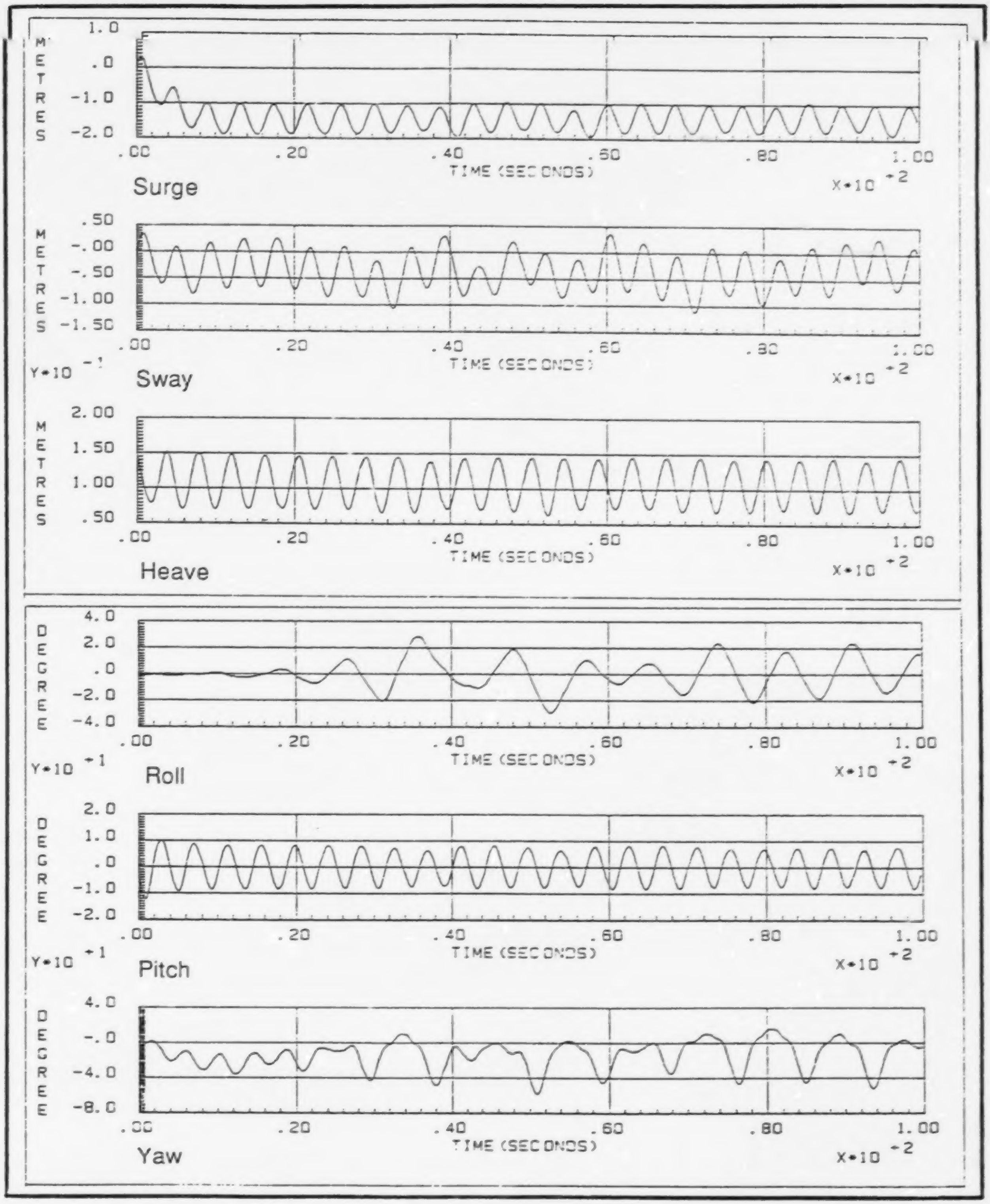
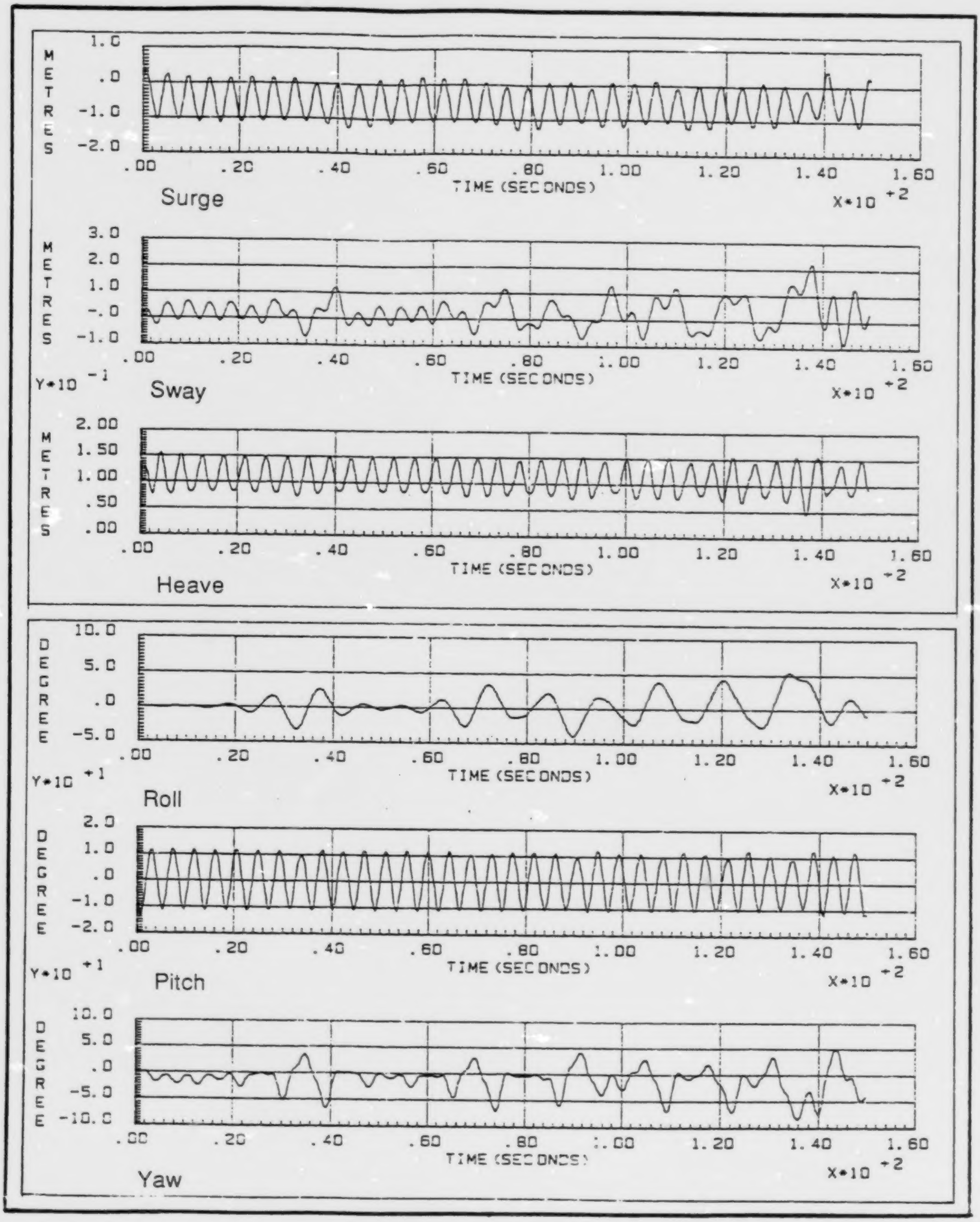


Figure 4.6 Motion History of "Hard Chine" Vessel in Case WM4-5%



Surge

Sway

Heave

Roll

Pitch

Yaw

Figure 4.7 Motion History of "Round Bilge" Vessel in Case EM4

5. DYNAMIC BEHAVIOUR AT RESONANCE FREQUENCY

5.1 General

The dynamic time-history simulations for both fishing vessels were performed using the "Servo Design 2" with zero diffraction forces. For all the simulations, the speeds of the vessels at different wave headings were adjusted to yield an encountering frequency equal to their natural roll frequency. For the "hard chine" vessel, the simulation exercises were done using 0.4 % critical roll damping as well as a 10 % critical in order to assess the damping effect.

Based on the analysis results of the "hard chine" vessel, two critical scenarios were identified and used for subsequent analysis of the "round bilge" vessel.

The complete simulation matrix is given as follows:

HARD CHINE VESSEL :

Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
WSM1	5.0	1/12	0.4
WSM2	30.0	1/12	0.4
WSM3	50.0	1/12	0.4
WSM4	5.0	1/7	0.4
WSM5	30.0	1/7	0.4
WSM6	50.0	1/7	0.4
WSM2-10%	30.0	1/12	0.4
WSM3-10%	50.0	1/12	0.4
WSM4-10%	5.0	1/7	0.4
WSM5-10%	30.0	1/7	0.4
WSM6-10%	50.0	1/7	0.4

ROUND BILGE VESSEL :

Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
ESM2-10%	30.0	1/12	10.0
ESM5-10%	30.0	1/7	10.0
ESM6-10%	50.0	1/7	10.0

All runs were made for 400 time steps at a time increment of 0.25 s .

5.2 Description of Capsize Mechanism

With the encountering frequency equal to the vessel's natural frequency, the large roll motions are essentially attributable to the roll resonance. The roll resonant motion builds up rapidly and capsizing occurs as the vessel rolls into a wave crest. The essential cause for this mode of capsize is due to the stability reduction in waves, which becomes the greatest in a wave crest amidships.

However, the resonant motion is sensitive to the damping of the vessel. Increases in damping can reduce the amplitude of the vessel motion and in turn improve the survivability in heavy seas, through the presence of greater dynamic stability at smaller angles.

5.3 Results Summary

The graphical outputs of the motion time-history plots of all simulation runs specified above are provided in Appendix E. The results summary is presented in Figure 5.1.

With a 0.4 % of critical damping in roll, the "hard chine" vessel capsized in all wave conditions, except the case WSM1 in which the vessel remained stable in a 1.911 metre high, 5 ° stern wave.

With the roll damping increased to 10 % critical, the vessel survived in most conditions, with the exception of the two high sea conditions, WSM5-10% and WSM6-10% , with the waves advancing in the 30 ° and 50 ° directions respectively.

As for the "round bilge" vessel, it survived in both the latter two critical high sea conditions.

5.4 Discussions of Results

For the "hard chine" vessel :

In all capsized cases with a 0.4 % critical roll damping, the "hard chine" vessel usually experienced a rapid build-up in roll motions in about 2 to 3 cycles, culminating in a capsize (see Figure 5.2 for case WSM5). Typical pitch angles were about plus and minus 10 degrees.

With the roll damping increased to 10 % critical, the dynamic stability of the vessel was somewhat improved with capsizing occuring only in the steep wave and the near quartering seas (cases WSM5-10% and WSM6-10%). Shown in Figures 5.3 and 5.4 are the motion time histories of the two capsized cases. Again the rolling motions in both cases were similar to the previous 0.4 % damping

WAVE HEIGHT \ DIRECTION	5^0	30^0	50^0
1/12L=1.911m	STABLE	CAPSIZE WSM2	CAPSIZE WSM3
1/7L=3.24m	CAPSIZE WSM4	CAPSIZE WSM5	CAPSIZE WSM6

HARD CHINE VESSEL

0.4% Critical Roll Damping

WAVE HEIGHT \ DIRECTION	5^0	30^0	50^0
1/12L=1.911m	NOT REQUIRED	STABLE WSM2-10%	STABLE WSM3-10%
1/7L=3.24m	STABLE WSM4-10%	CAPSIZE WSM5-10%	CAPSIZE WSM6-10%

HARD CHINE VESSEL

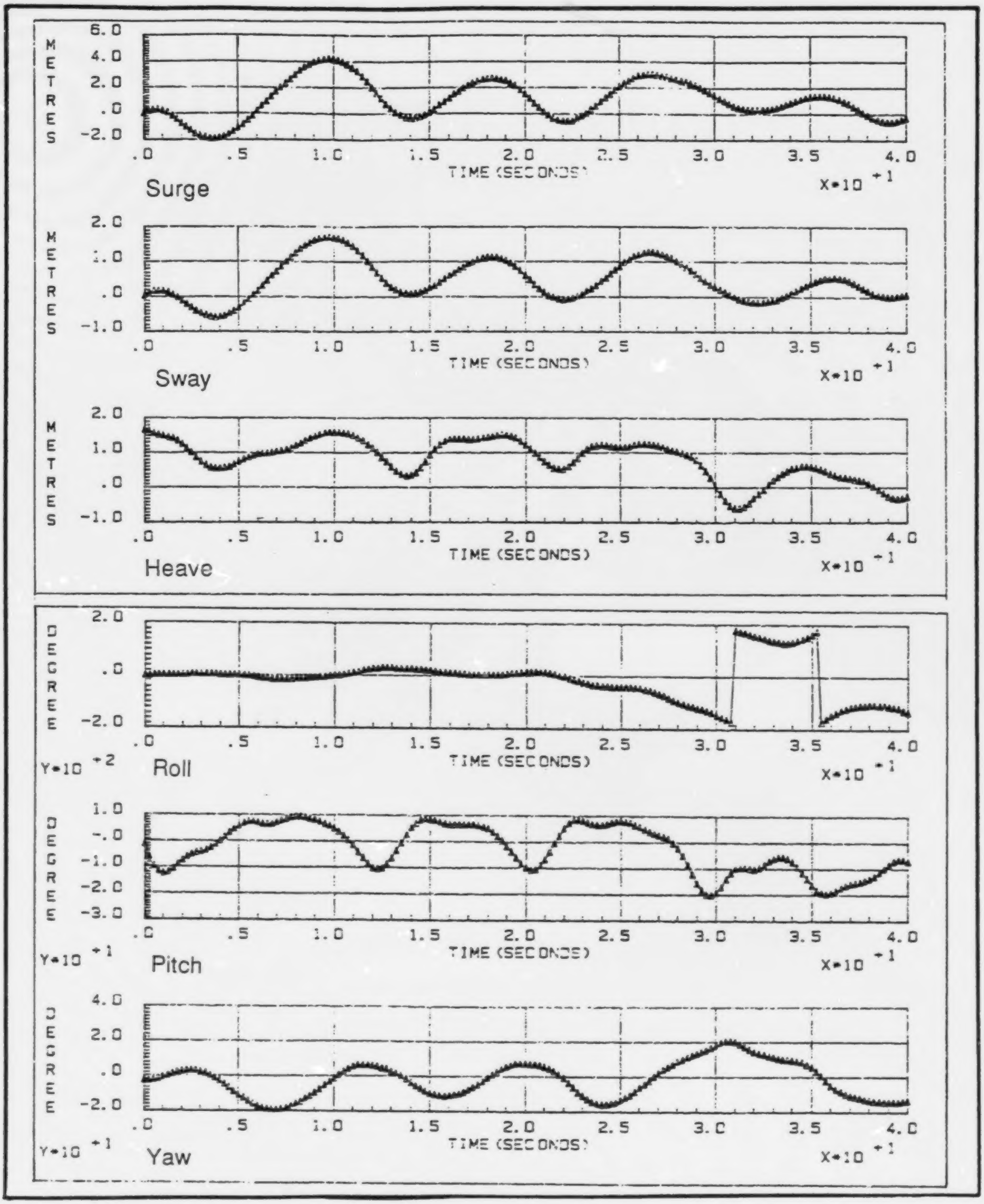
10% Critical Roll Damping

WAVE HEIGHT \ DIRECTION	5^0	30^0	50^0
1/12L=1.911m	NOT REQUIRED	STABLE ESM2-10%	NOT REQUIRED
1/7L=3.24m	NOT REQUIRED	STABLE ESM5-10%	STABLE ESM6-10%

ROUND BILGE VESSEL

10% Critical Roll Damping

Figure 5.1 Results Summary at Resonance Frequency


Surge

Sway

Heave

Roll

Pitch

Yaw

Figure 5.2 Motion History of "Hard Chine" Vessel in Case WSM5

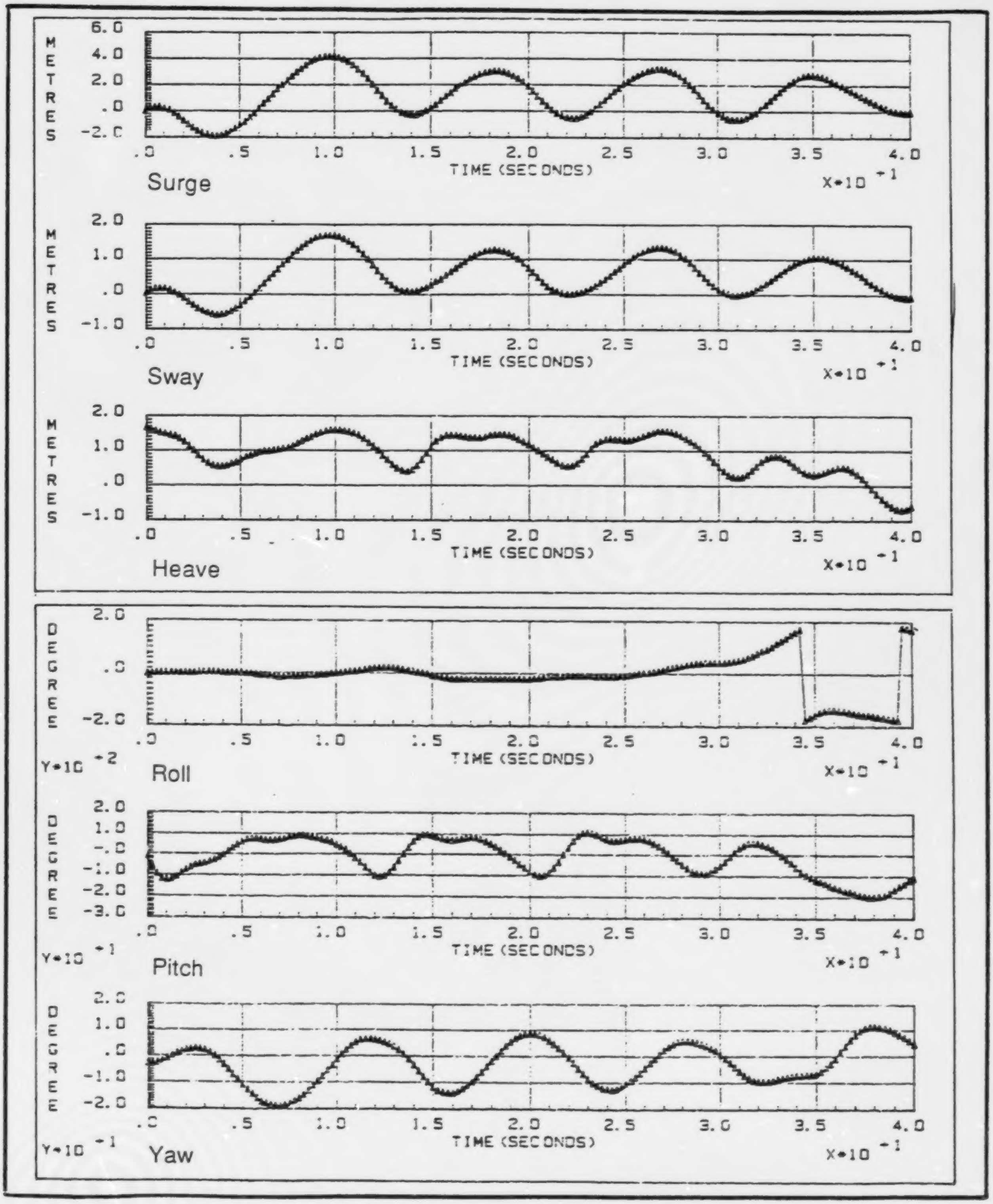


Figure 5.3 Motion History of "Hard Chine" Vessel in Case WSM5-10%

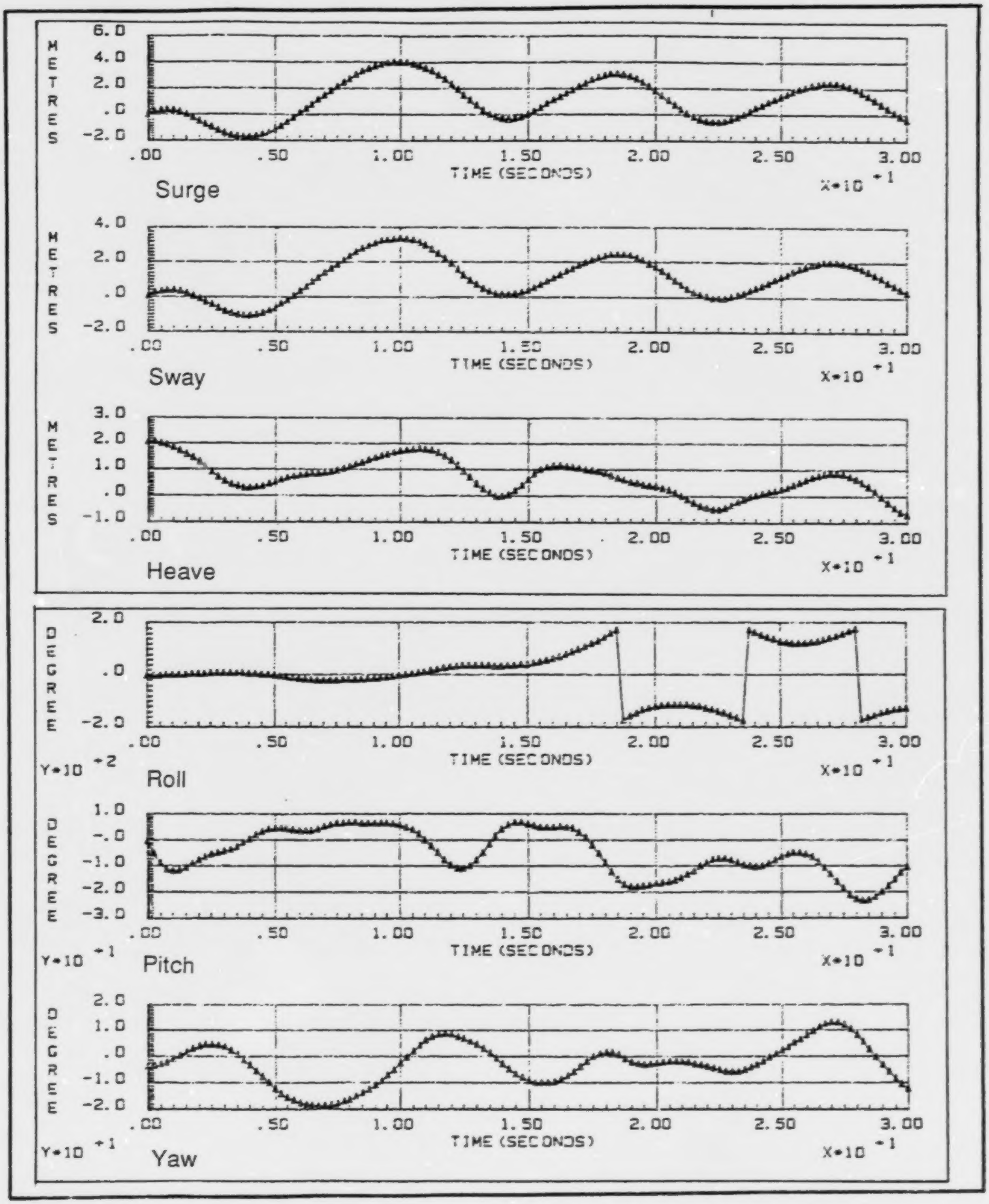


Figure 5.4 Motion History of "Hard Chine" Vessel in Case WSM6-10%

capsized cases. The roll motions built up rapidly to about 30 ° in 2 to 3 cycles (28 s in Figure 5.3 and 15 s in Figure 5.4), and the vessel capsized as the last wave crest moved in amidships.

The roll motion, velocity and acceleration time histories, prior to the capsize, for the case WSM5-10% are given in Figure 5.5, and the hydrostatic, Froude-Krylov, and total moments in roll are shown in Figure 5.6. As can be seen from Figure 5.5, the roll motions built up in about 3 cycles. The trouble appears to begin at the end of the second cycle when the vessel was on the front slope of the wave (16 s). The vessel rolled to the port to a maximum angle of 20 °, and should begin to heel over to the other side as the wave crest moved in (16 -18 s). Rather, the large overturning moment (Froude - Krylov) worked against a diminishing restoring moment (hydrostatic) during this period in upholding the vessel from righting (Figure 5.6). At 20 s the vessel is on the back slope of the third wave (crest at 0.25L forward) and the dynamic wave force (Froude-Krylov) started to upright the vessel. As the wave passed the vessel, it rolled further towards the starboard, passing the upright position (26 s) and acquiring high angular speed. By the time the last wave moved in (28 s), the vessel acquired sufficient angular momentum to carry it beyond the critical angle of zero righting arm; and as the wave crest passed the vessel the dynamic wave force accelerated further over, leading to the capsize. Figure 5.7 shows the sequences of the capsize.

The reduction in the number of capsizes in the 10 % damping case clearly demonstrates the effect of damping on the roll motions. It appears that the vessel becomes dynamically more stable as the damping increases.

For the "round bilge" vessel :

The two wave conditions indicated as critical to the "hard chine" vessel were applied to the "round bilge" vessel (case ESM5-10% and ESM6-10%). The dynamic motion characteristics of these two cases are given in Figures 5.8 and 5.9. Under both wave conditions, the vessel with a 10 % critical roll damping experienced no danger of capsizing at all.

The significant peak roll angles predicted in the following 30 ° sea (ESM5-10%) and the following 50 ° sea (ESM6-10%) were 18 ° and 26 ° respectively, to the starboard side. Typical pitch angles were plus and minus 10° in both cases.

Shown in Figure 5.10 is the enlarged view of the initial transient (first 128 records) of the roll motion, velocity and acceleration time histories for the case ESM5-10%. It can be seen that the appearance of the roll motion record resembles a classical resonant oscillation being excited at its natural frequency. The roll peaks are exactly 8.5 s apart. The roll motion grew to the maximum amplitude very rapidly. Shown in Figure 5.10, the vessel took a maximum roll on the crest (12 s), then "snapped" back upright and heeled over to the other side as the trough moved to amidships (16 s). This process continued as the waves

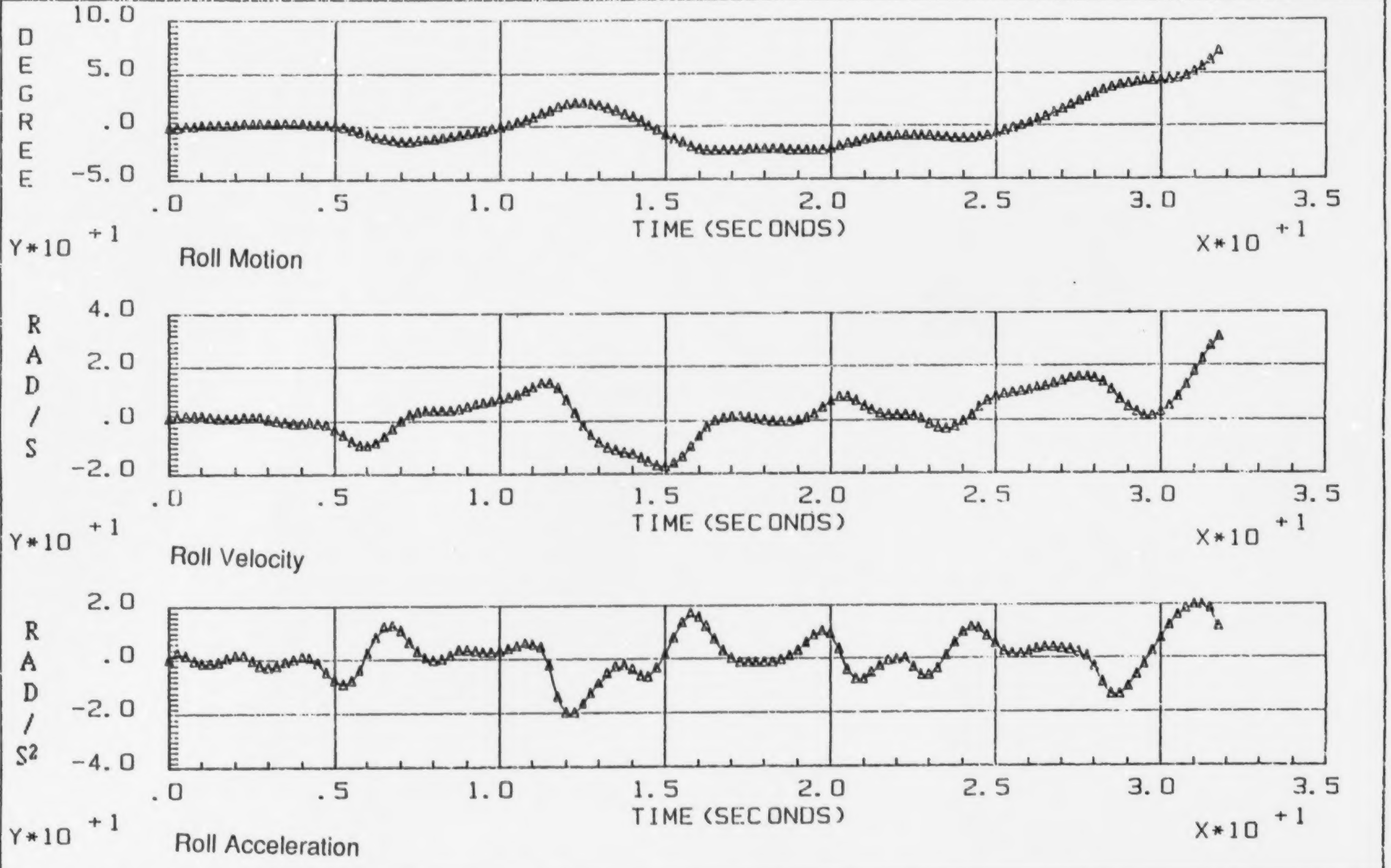
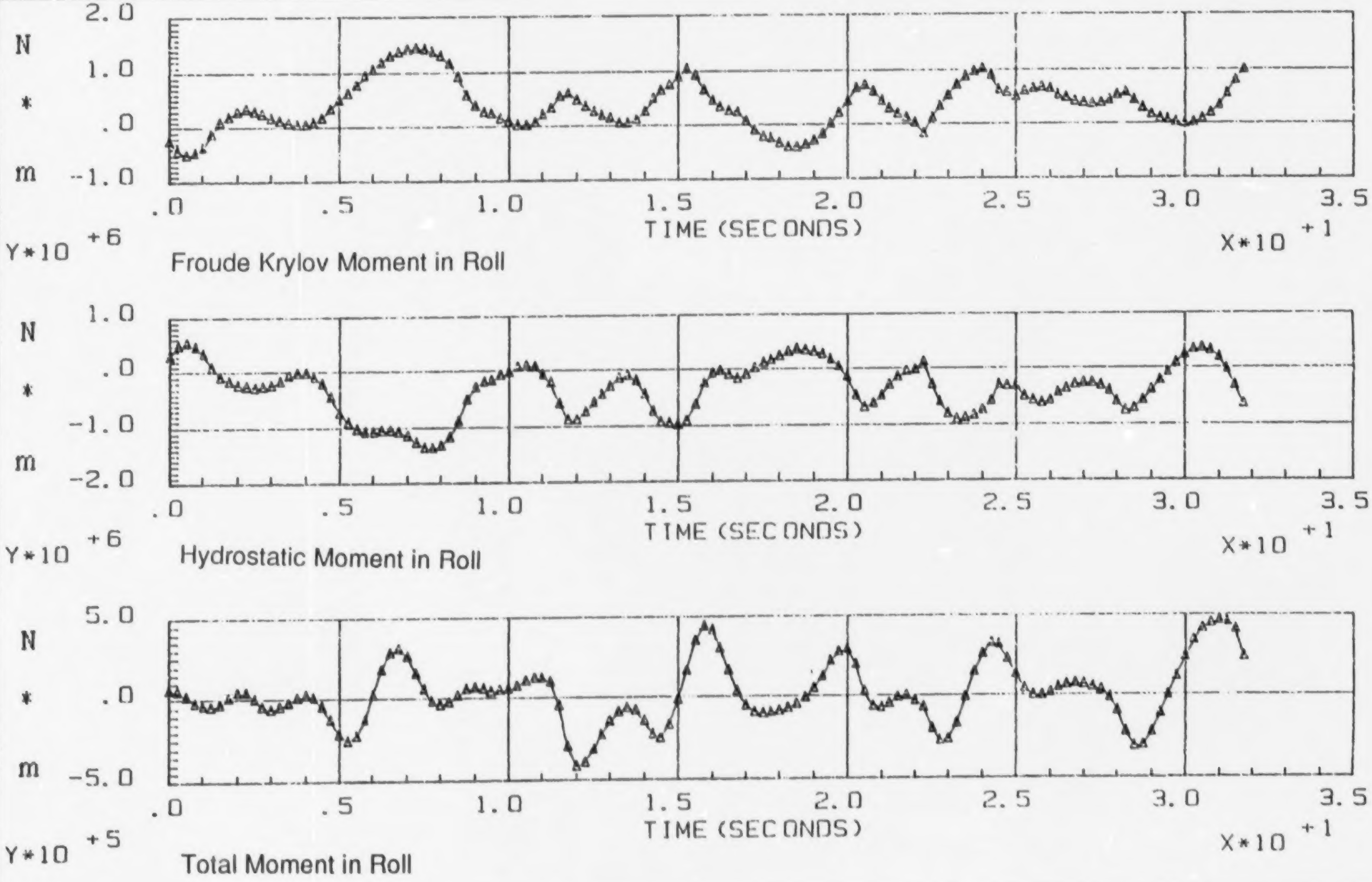


Figure 5.5 Roll Motion, Velocity and Acceleration for Case WSM5-10%



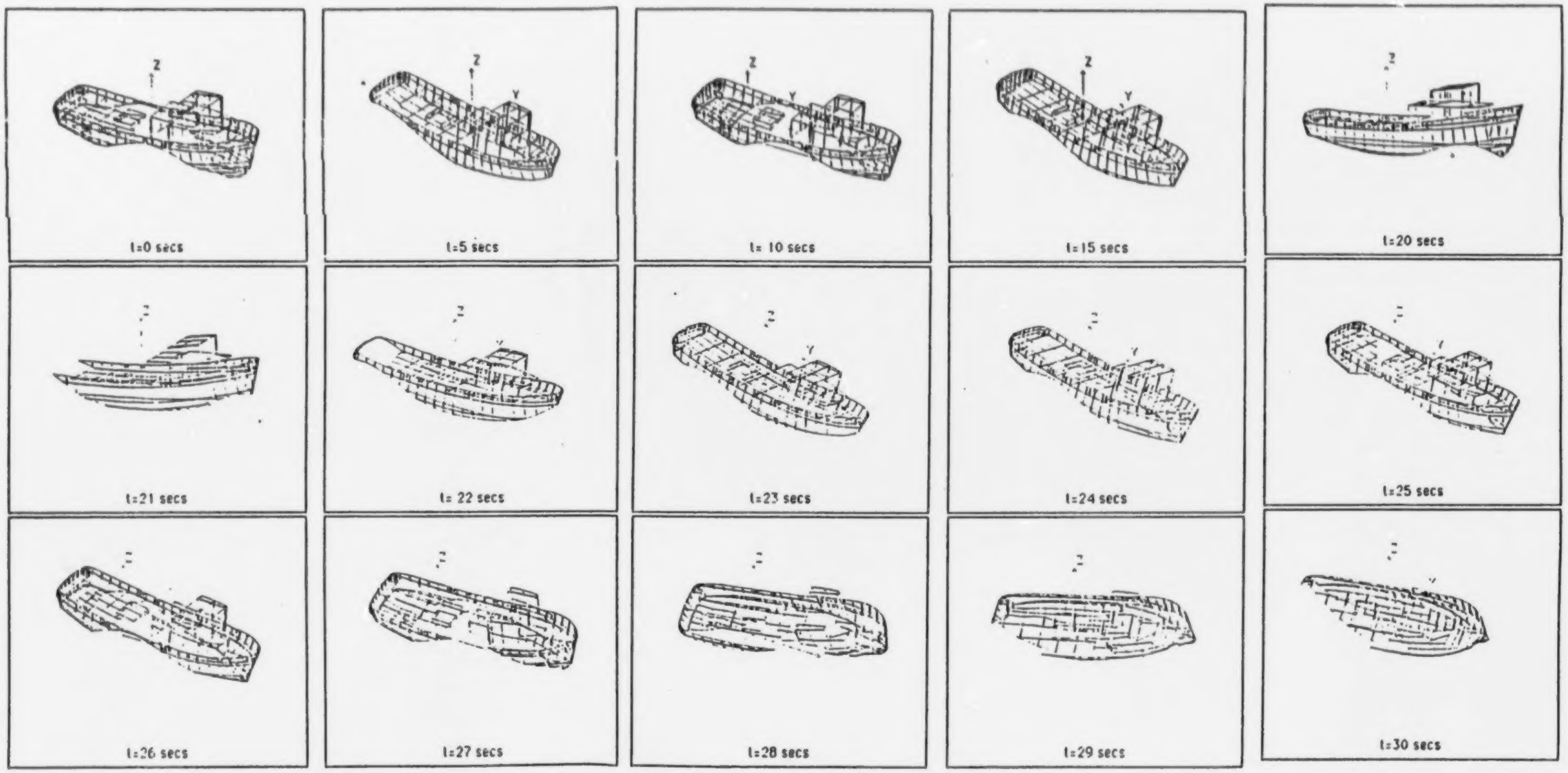
Froude Krylov Moment in Roll

Hydrostatic Moment in Roll

Total Moment in Roll

Figure 5.6 Roll Moments for Case WSM5-10%

Figure 5.7 Capsize Sequences of "Hard Chine" Vessel in Case WSM5-10%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Y
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t=5 secs
t= 10 secs
t=15 secs
t=20 secs
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t=30 secs

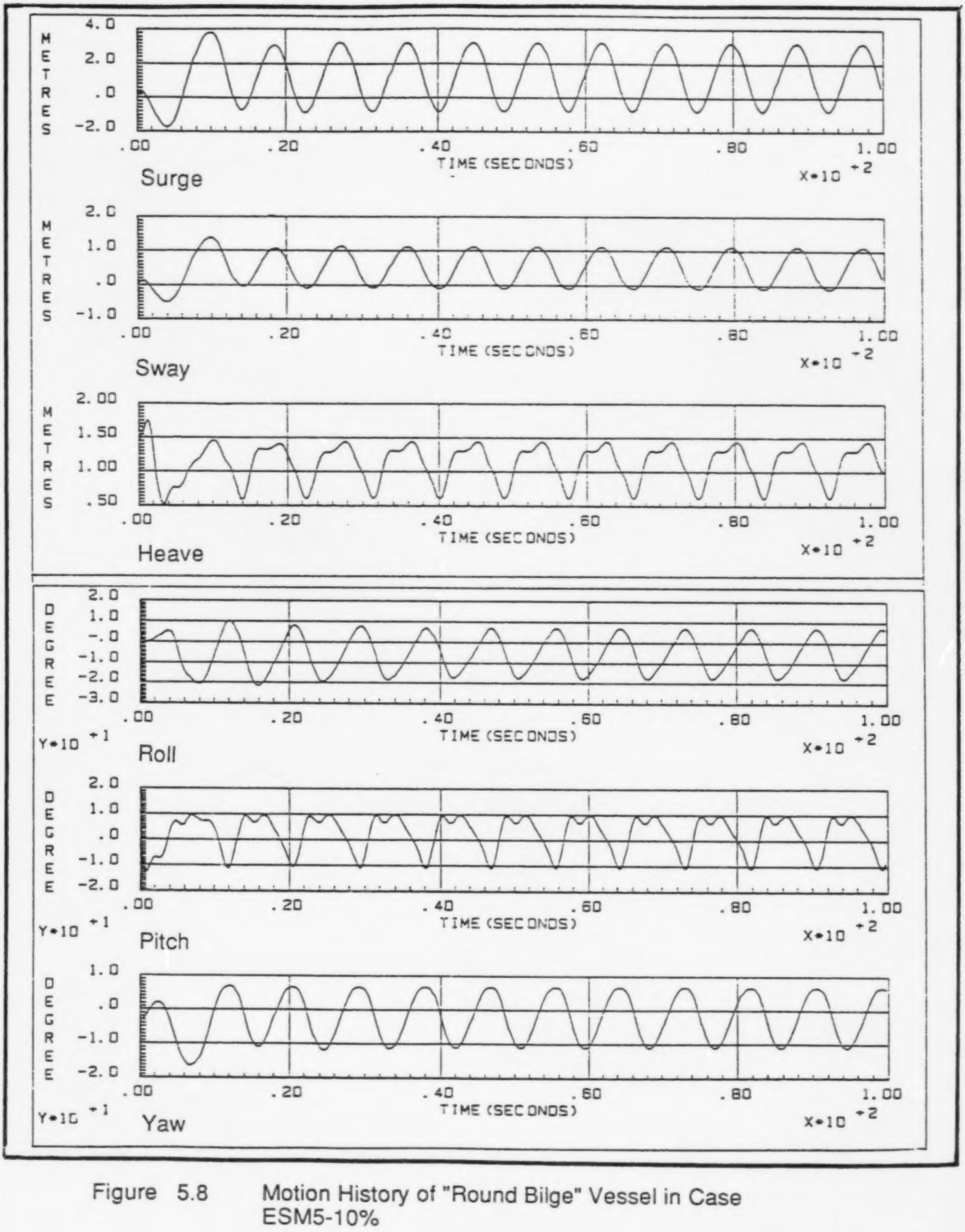


Figure 5.8 Motion History of "Round Bilge" Vessel in Case ESM5-10%

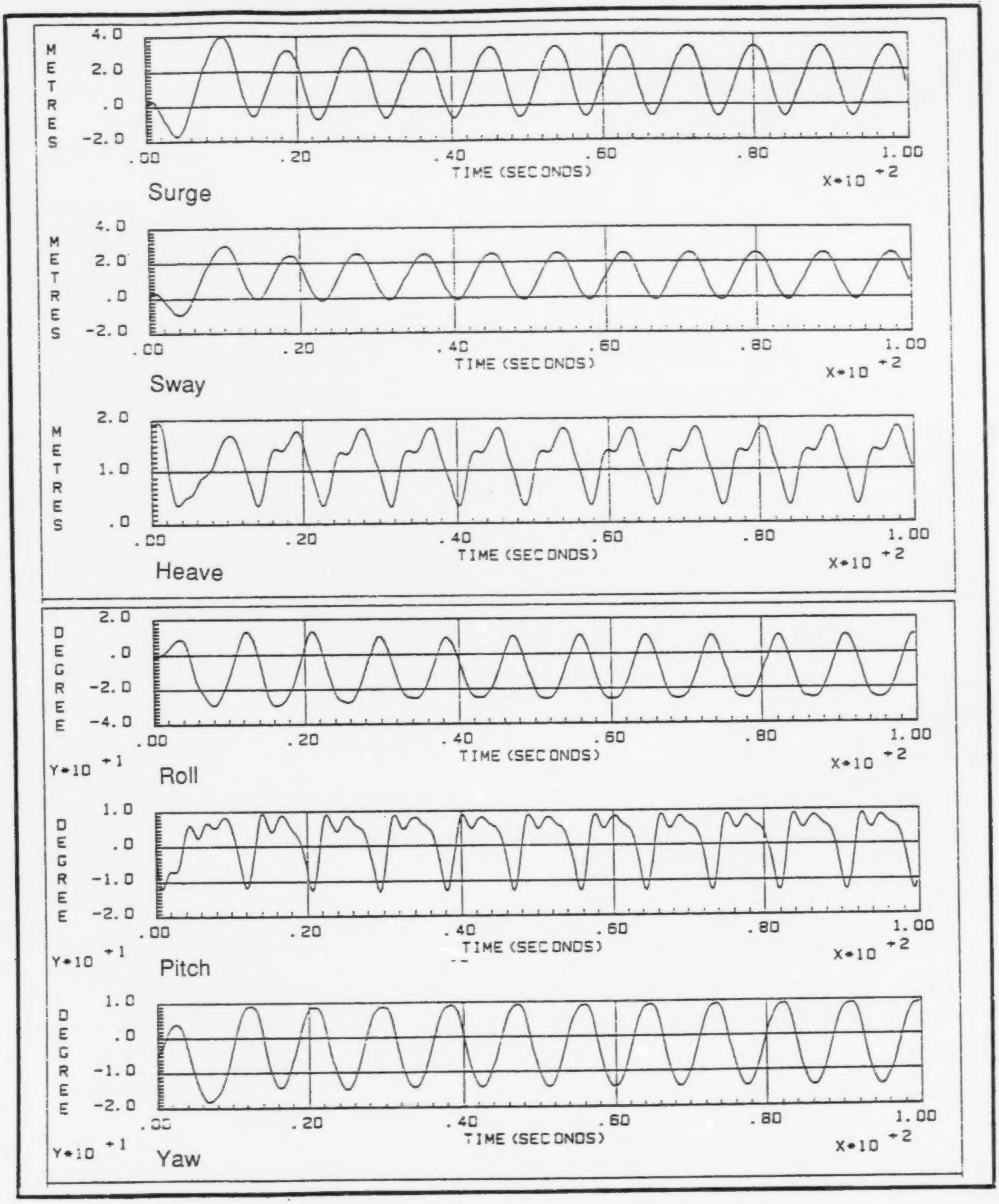
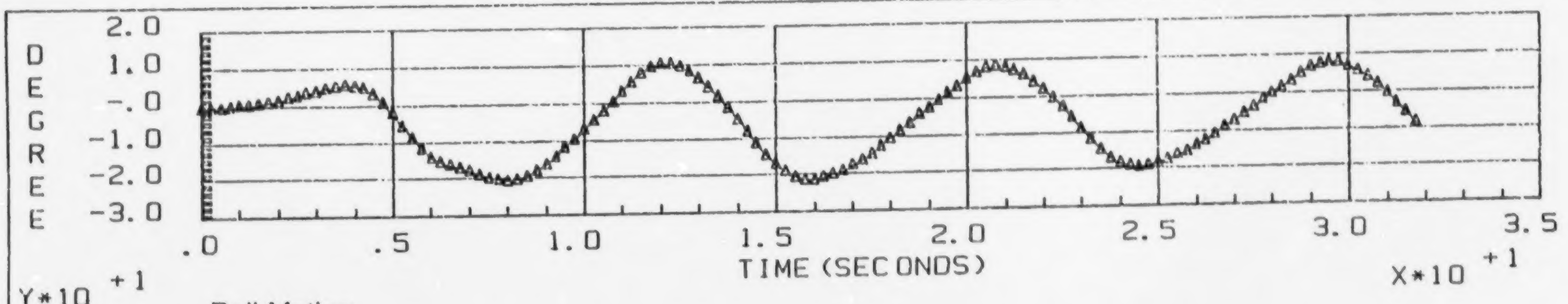
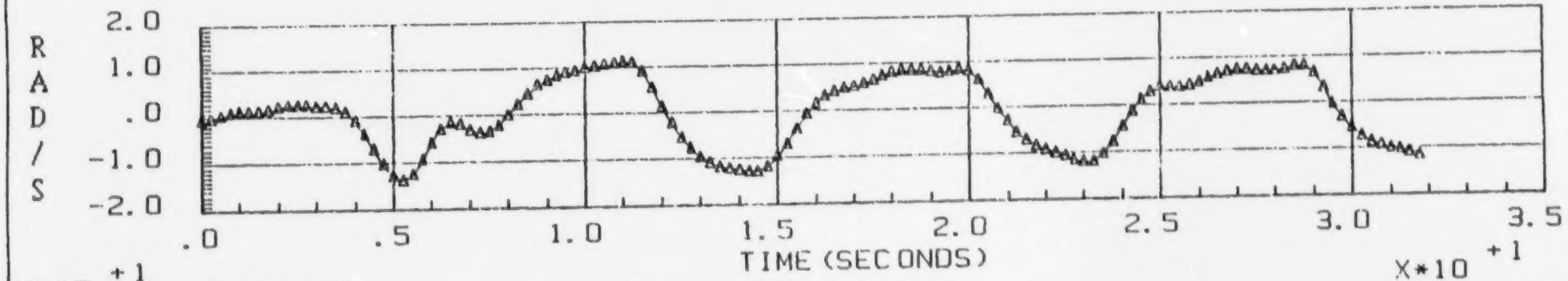


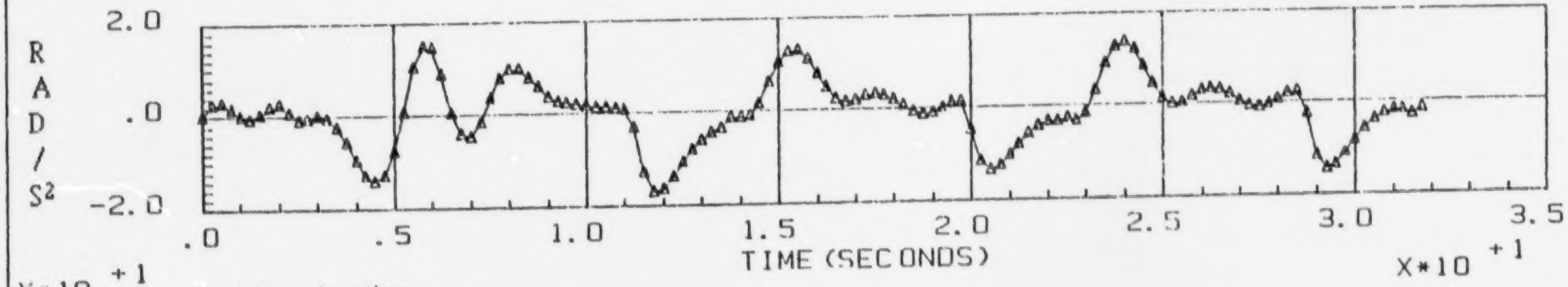
Figure 5.9 Motion History of "Round Bilge" Vessel in Case ESM6-10%



Roll Motion



Roll Velocity



Roll Acceleration

Figure 5.10 Initial Transient Roll Motion, Velocity and Acceleration for Case ESM5-10%

passed the vessel. The roll motion, however, did not build up further to larger angles, indicating that the vessel always had sufficient righting arms (GZ) to restore its dynamic equilibrium position.

5.5 Comparison Between Vessels

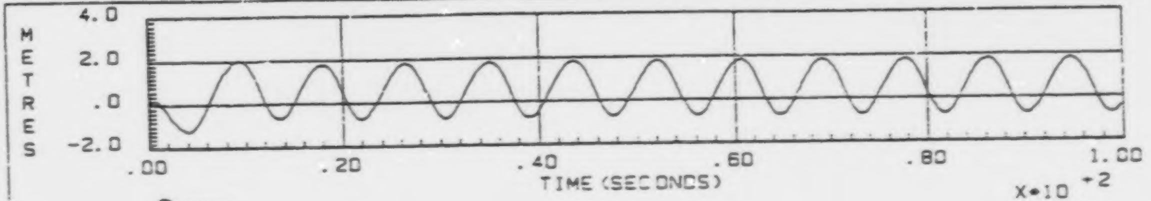
From the above discussion, it appears that the "round bilge" vessel is dynamically better than the "hard chine" vessel. However, since the roll motions for both vessels normally did not exceed 20 ° during the build-up, even in the steep waves, the difference in the large angle stability cannot possibly be the answer to the difference in performance between the vessels.

In an attempt to seek an explanation, further comparison was made for a moderate sea state (1/12L), where no capsize is taking place. In this case the simulation runs WSM2-10% and ESM2-10% were chosen and their motion records are given in Figures 5.11 and 5.12 respectively. It can be seen that the motions of both vessels are essentially the same. The steady roll angles for the "hard chine" vessel are plus and minus 13 ° with 3 ° more roll on the port side (oscillating from 10 ° starboard to 16 ° port). As for the "round bilge" vessel the roll amplitude is 7.75 ° with 3.25 ° more on the port side (oscillation between 4.5 ° starboard and 11 ° port).

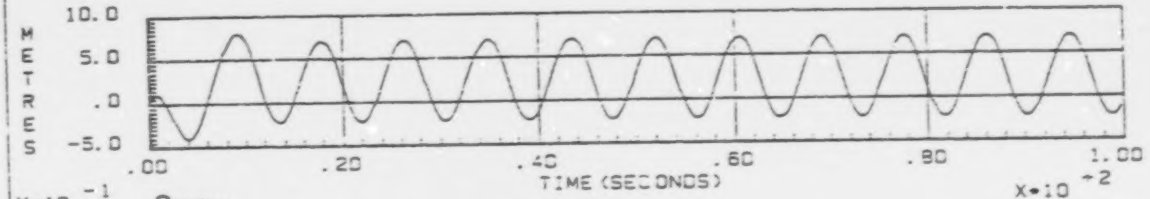
To examine why the "round bilge" vessel rolls less than the "hard chine" vessel, "quasi-static inclining" computer runs were made for both vessels: (1) with the wave trough at amidships (at 81.5 s), (2) with the crest at .25L forward (at 83.75 s), and (3) with crest at amidships (at 86.25 s). The results are given in Figures 5.13 and 5.14. A direct comparison of these "quasi-static" GZ curves is shown in Figure 5.15.

It can be seen that the GZ curves for both vessels on the first two wave positions (Figure 5.15 a and b) are effectively the same for roll angles up to 25 °, but with some noticable differences beyond that angle.

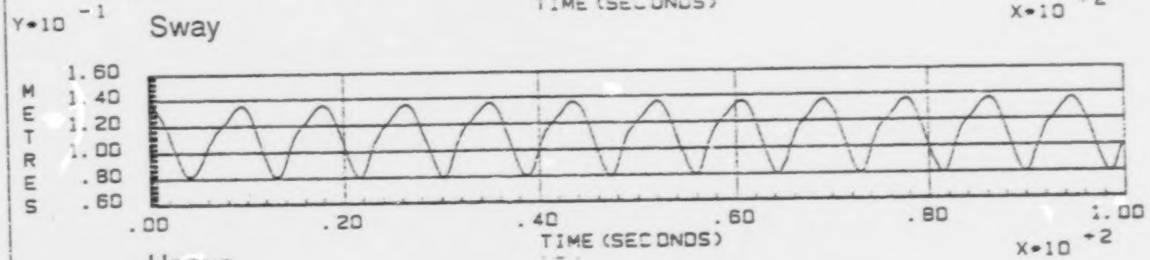
In contrast, the GZ curves for the wave crest condition (Figure 5.15 c) show striking differences between the two vessels. The righting arms of the "hard chine" vessel peaks at 10 ° and vanishes at about 27 °, while for the "round bilge" vessel the GZ curve reaches the maximum at 28 ° and continues to be positive beyond 40 °. The dynamic stability of the vessels up to 27 °, assessed by the areas under the GZ curves indicate that the "round bilge" vessel is superior to the "hard chine" vessel. Noting that the GZ curves below the angle of deck edge immersion are essentially influenced by the underwater hull form, the marked differences in GZ values and hence dynamic stability of the two vessels in angles between 6 ° and 27 ° are obviously brought about by their difference in hull design.



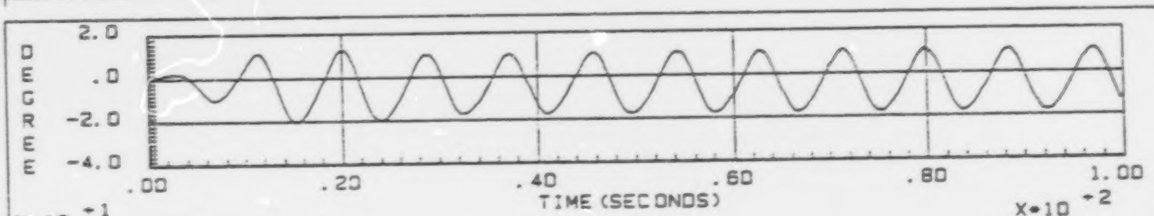
Surge



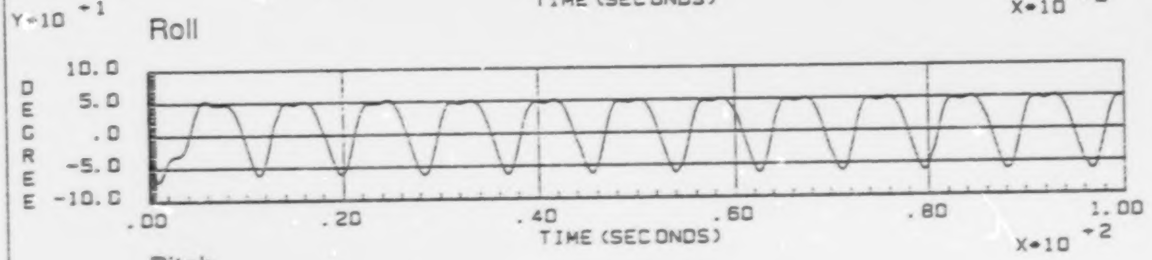
Sway



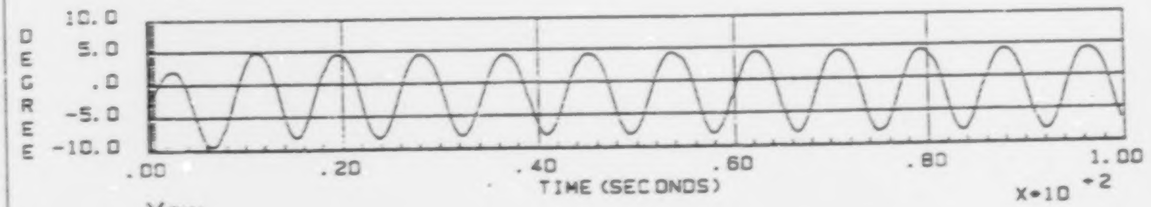
Heave



Roll



Pitch



Yaw

Figure 5.11 Motion History of "Hard Chine" Vessel in Case WSM2-10%

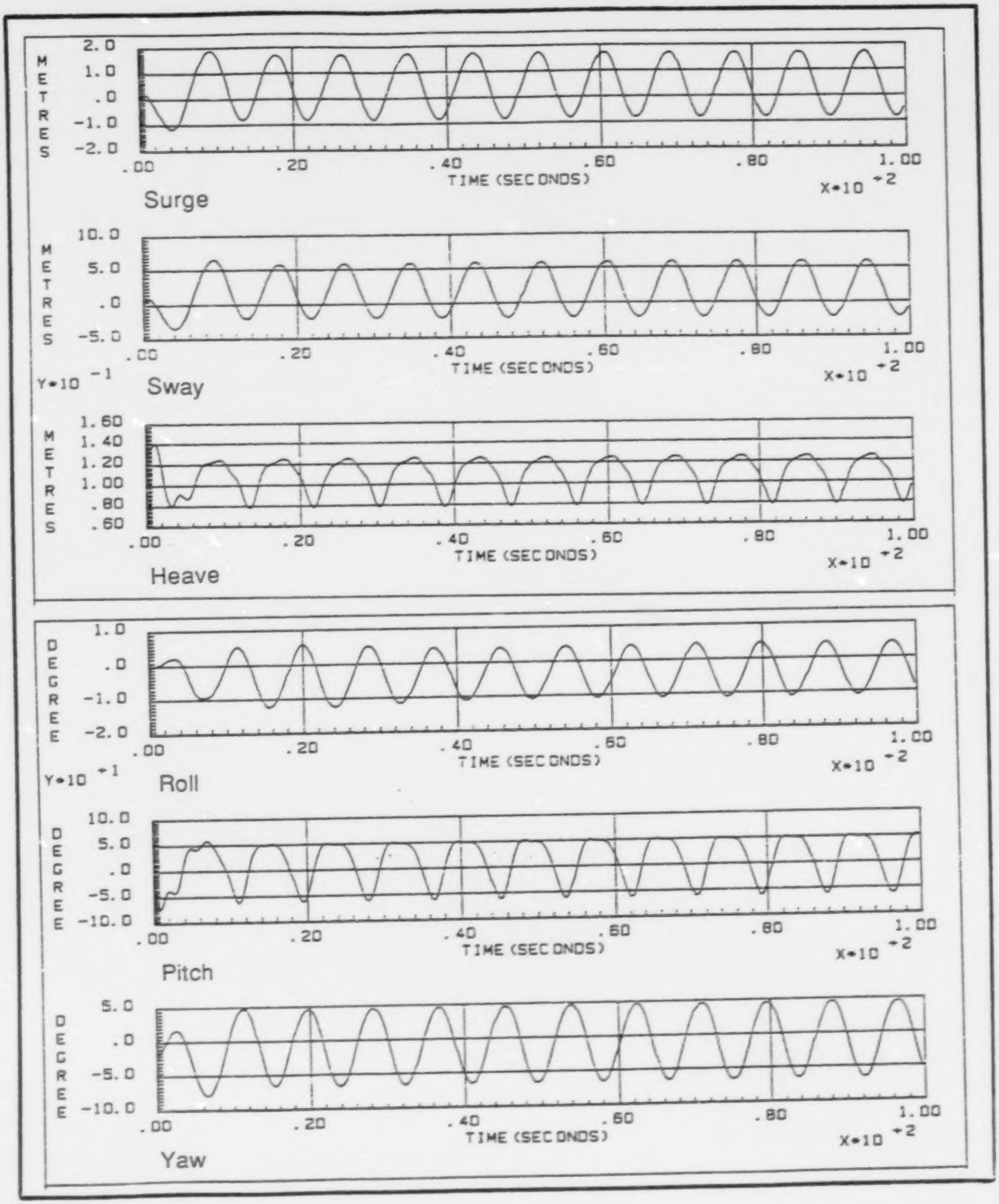


Figure 5.12 Motion History of "Round Bilge" Vessel in Case ESM2-10%

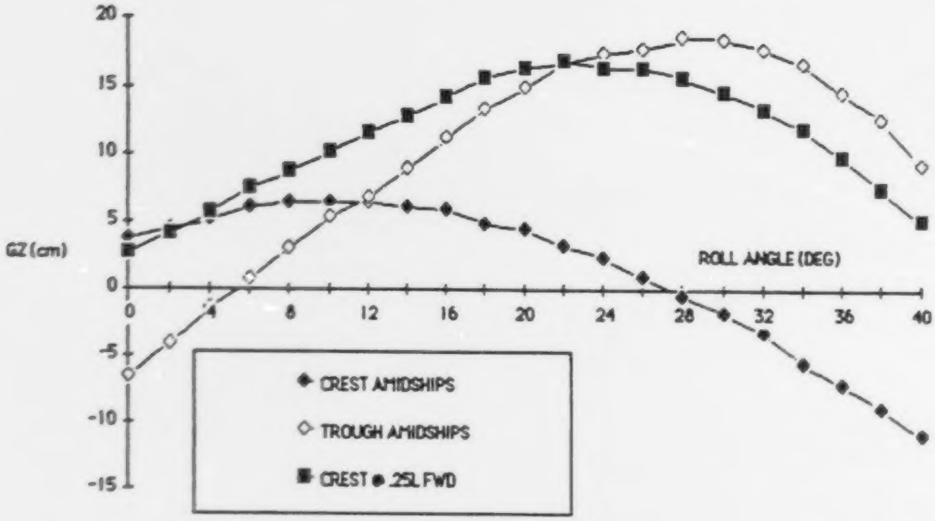


Figure 5.13 GZ Curves of "Hard Chine" Vessel in Waves

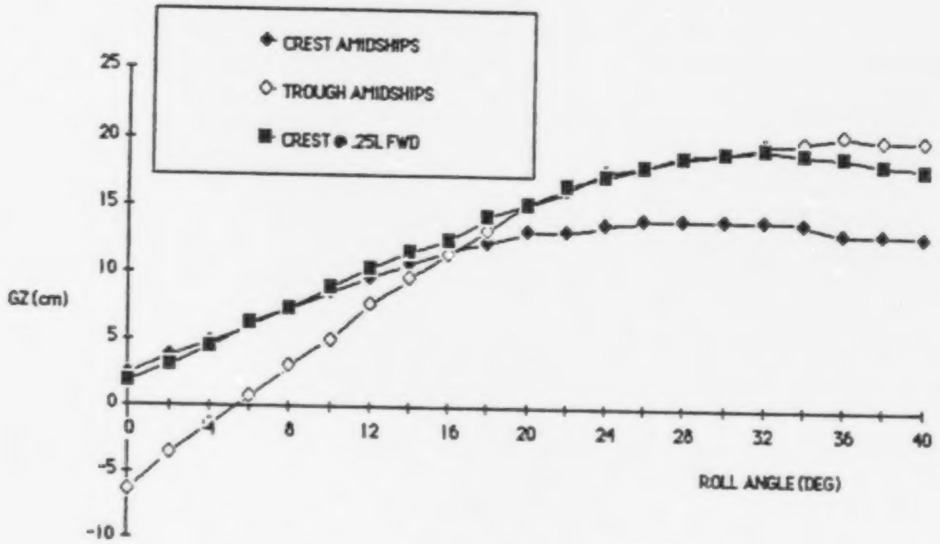
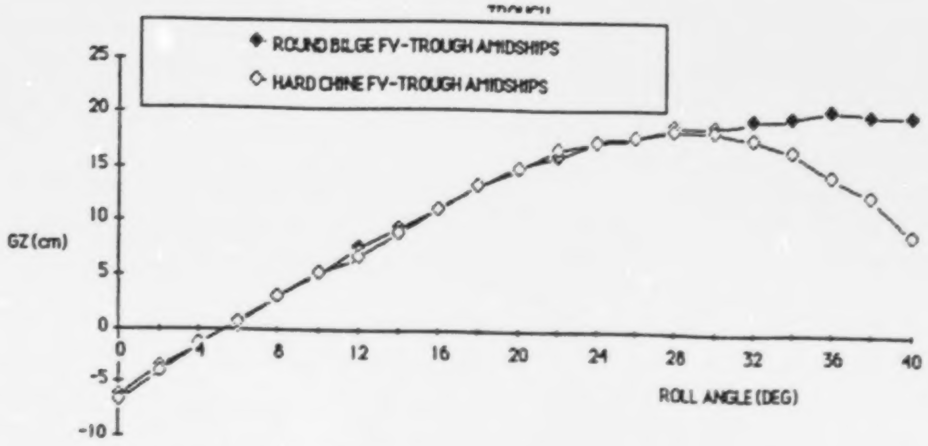
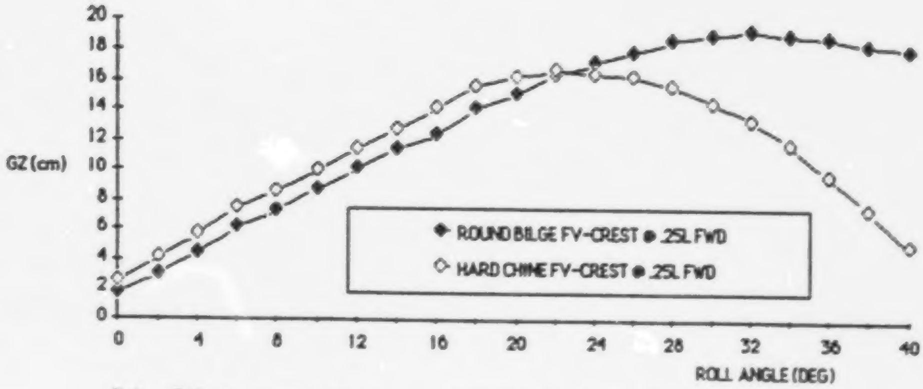


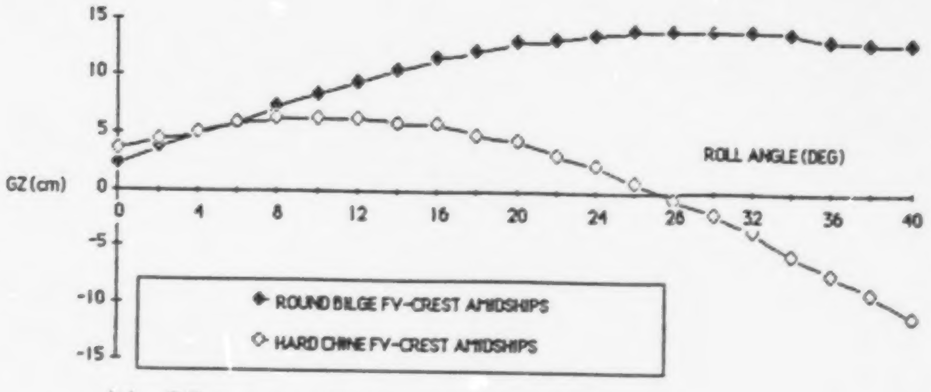
Figure 5.14 GZ Curves of "Round Bilge" Vessel in Waves



(a) GZ curves with wave trough amidships



(b) GZ curves with wave crest @ 0.25L forward



(c) GZ curves with wave crest amidships

Figure 5.15 Comparison of GZ Curves Between Vessels in Waves

6. DYNAMIC BEHAVIOUR OF THE "WAVE CREST CASE"

6.1 General

For these time history runs the vessel speed was adjusted to produce an encountering period of 100 s (near infinite). All runs were performed using the "Servo Design 2" and the diffraction forces were set equal to zero. For most runs 0.4% critical radiation roll damping as predicted by AQWA-LINE was used and a level of 10 % was used in two critical cases for assessing the damping effect.

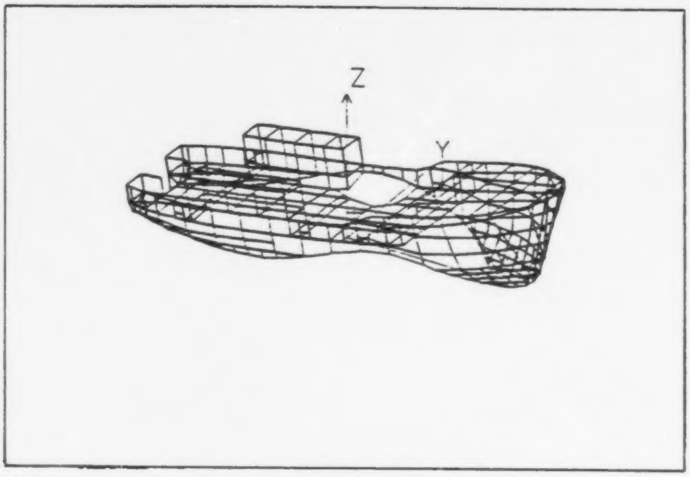
The initial start time was 0 s for all runs except ELS5, ELS6, WLS5 and WLS6. The former two runs were initially unstable when started at 0 s , as the transient forces caused the vessel to roll over immediately. A start time of 10 s was then used. For the purpose of performance comparison, the latter two runs was also started at the same initial time.

It is noted that the dynamic stability is dependent on the initial conditions of the simulation. For a physical system that is statically or dynamically unstable at the start, unreasonably excessive motions may develop in a few seconds, resulting in an unrealistic simulation. This is believed to have taken place with the ELS5 and ELS6 cases. Therefore it was decided to start the simulations at 10 s, with the second wave crest poised slightly forward of amidships. Under this condition, the vessel which is initially stable, although not in a state of equilibrium, quickly reaches dynamic equilibrium and reacts accordingly with the wave actions (see results in Appendix F). Figure 6.1(a) shows the vessel position relative to the wave at the start of 10 s, while given in Figure 6.1(b) is the position if started at 0 s for ELS5.

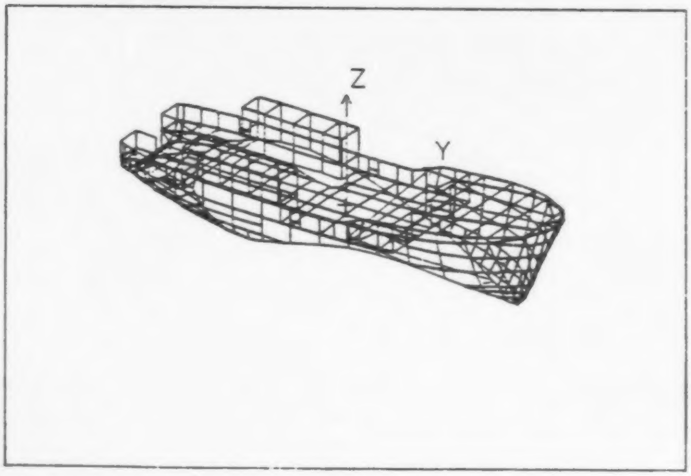
The following is a list of the test matrix with case numbers and corresponding wave conditions:

HARD CHINE VESSEL

Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
WLS1	5.0	1/12	0.4
WLS2	30.0	1/12	0.4
WLS3	50.0	1/12	0.4
WLS4	5.0	1/7	0.4
WLS5	30.0	1/7	0.4
WLS6	50.0	1/7	0.4
WLS4-10%	5.0	1/7	10.0
WLS5-10%	30.0	1/7	10.0



(a) Initial position at 10 seconds



(b) Initial position at 0 second

Figure 6.1 Vessel Positions at Start for Case ELS5

ROUND BILGE VESSEL

Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
ELS4	5.0	1/7	0.4
ELS5	30.0	1/7	0.4
ELS6	50.0	1/7	0.4

Also a number of "static" inclination tests were performed on both vessels where appropriate.

6.2 Description of Capsize Mechanism

The mechanism for this capsize is a sudden loss of stability resulting in roll over without any prior rolling motion. The requirements for this capsize to occur are:

a) a following sea.
b) the vessel travels with a forward speed close to the wave's forward velocity creating a very long encounter period.
c) a wave length about equal to the ship length.
d) a steep wave.

At the crest the vessel cannot produce enough hydrostatic righting moment to overcome the dynamic overturning wave forces. When the vessel is traveling near the wave speed, the vessel remains on the crest for a long period of time, permitting the net overturning moment to cause rollover.

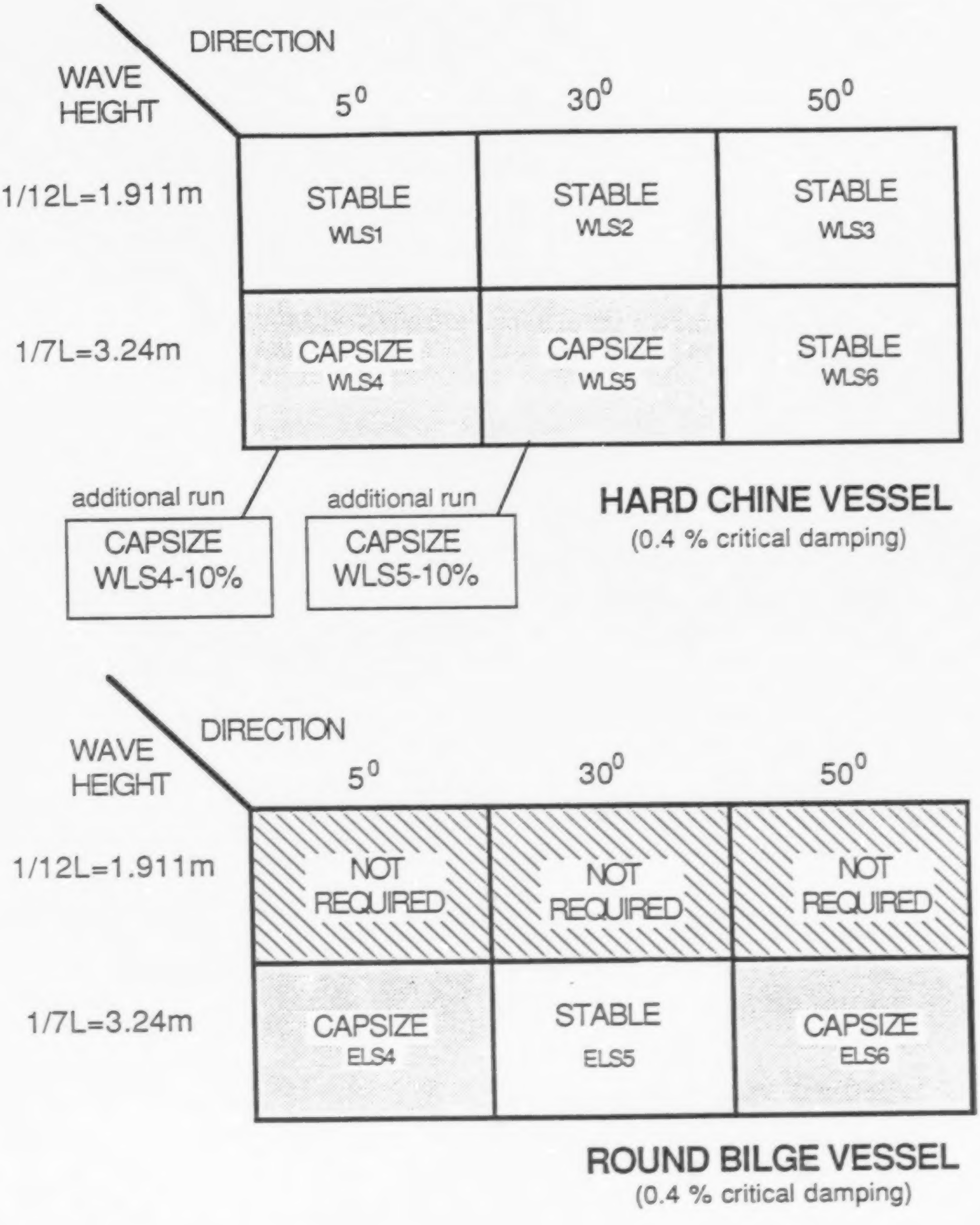
6.3 Results of "Wave Crest" Case

A summary of the results are shown in Figure 6.2 and the complete set of motion time histories are given in Appendix F.

For the "hard chine" vessel at the three wave directions and at the lower wave height (1/12 L) no capsizes occurred and all roll angles were 20 ° or less. For the larger wave height (1/7 L) capsize occurred at wave directions of 5 ° and 30 °.

For the "round bilge" vessel three runs were performed at the higher wave height. The vessel capsized in the 5 ° and 50 ° wave directions.

Note that these runs were carried out using 0.4 % critical damping.



WAVE HEIGHT \ DIRECTION	5^0	30^0	50^0
1/12L=1.911m	STABLE WLS1	STABLE WLS2	STABLE WLS3
1/7L=3.24m	CAPSIZE WLS4	CAPSIZE WLS5	STABLE WLS6

additional run: CAPSIZE WLS4-10%

additional run: CAPSIZE WLS5-10%

HARD CHINE VESSEL
(0.4 % critical damping)

WAVE HEIGHT \ DIRECTION	5^0	30^0	50^0
1/12L=1.911m	NOT REQUIRED	NOT REQUIRED	NOT REQUIRED
1/7L=3.24m	CAPSIZE ELS4	STABLE ELS5	CAPSIZE ELS6

ROUND BILGE VESSEL
(0.4 % critical damping)

Figure 6.2 Results Summary for "Wave Crest" Case

6.4 Discussion of Results

The time histories for the steep following 5 ° sea (case WLS4) as given in Figure 6.3 shows the classical case of sudden loss of stability and will be discussed below.

At the start of the time history (0 s) the vessel is perched on the crest of a wave and is given an initial position calculated from linear wave theory. The wave forward velocity is slightly greater than the vessel's and thus the waves slowly overtake the vessel.

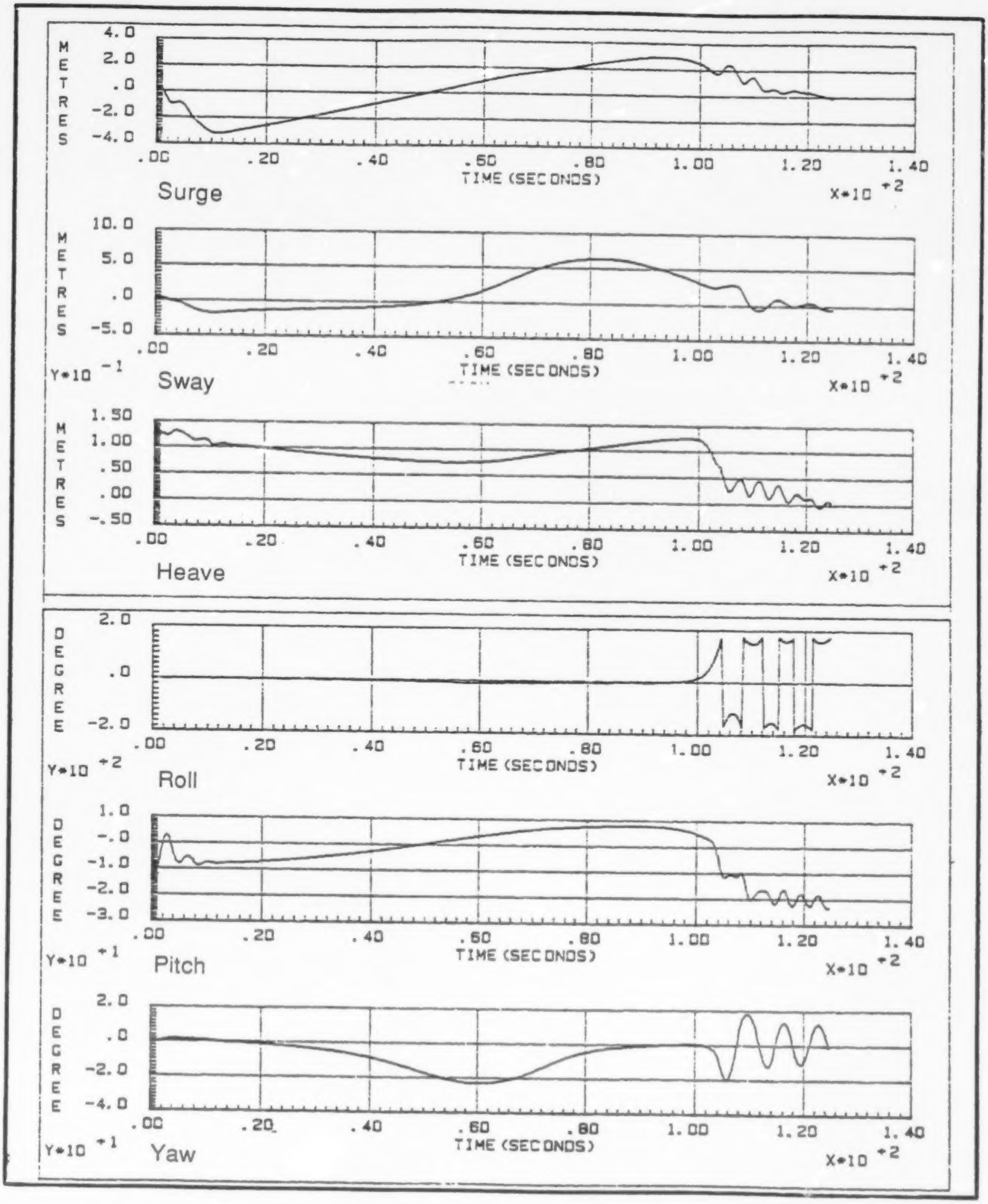
Between 0-12 s the vessel goes through a transient period until the vessel reaches dynamic equilibrium. At time equal to 25 s the vessel is on the back of a wave. At time equal to 55 s the vessel is in the trough. At time equal to 75 s the vessel is on the front slope of a wave and at time equals 100 s the vessel is again on a wave crest.

As can be seen in Figure 6.3 at 80 s the vessel is upright on the wave front slope. As the wave approaches amidships (94 s) the vessel begins to roll to starboard (positive roll) and as the wave passes amidships the roll angle becomes larger, roll velocity increases, and the vessel rolls over. From 100 s to 104 s the roll angle goes from 10 deg to 180 ° (i.e. upside down). The sequences of the capsize is shown in Figure 6.4 at 20 s interval from 0 to 80 s and then every second from 95 s to 103 s.

Figure 6.5 shows the complete Froude Krylov, hydrostatic, and total moment time histories during a critical period from 96 s to 103 s. The positive overturning moment (Froude Krylov) works against a diminishing hydrostatic restoring moment, causing the vessel to roll more to starboard. Once past a roll angle of 43 ° at 103 s, the hydrostatic roll moment becomes overturning further accelerating the capsize.

6.5 "Quasi-Static" Inclining Tests

In order to study this loss of stability further, a number of "quasi- static inclining test" computer runs were performed for the "hard chine" vessel. First, an "inclining test" was performed for the vessel on a wave crest (1/7 L) using the x-y-z position at 99 s from WLS4, as the initial position. The vessel was then rotated about this initial position at increments of 2 ° to a maximum of 80 °. The Froude Krylov, hydrostatic, and total roll moments are shown in Figure 6.6(a) and the effective GZ curve obtained from dividing the total moment by the ship mass is shown in Figure 6.6(b). The results of this "quasi-static" run can be compared to the section of the dynamic run shown in Figure 6.5. It can be seen that they are essentially the same.



Surge

Sway

Heave

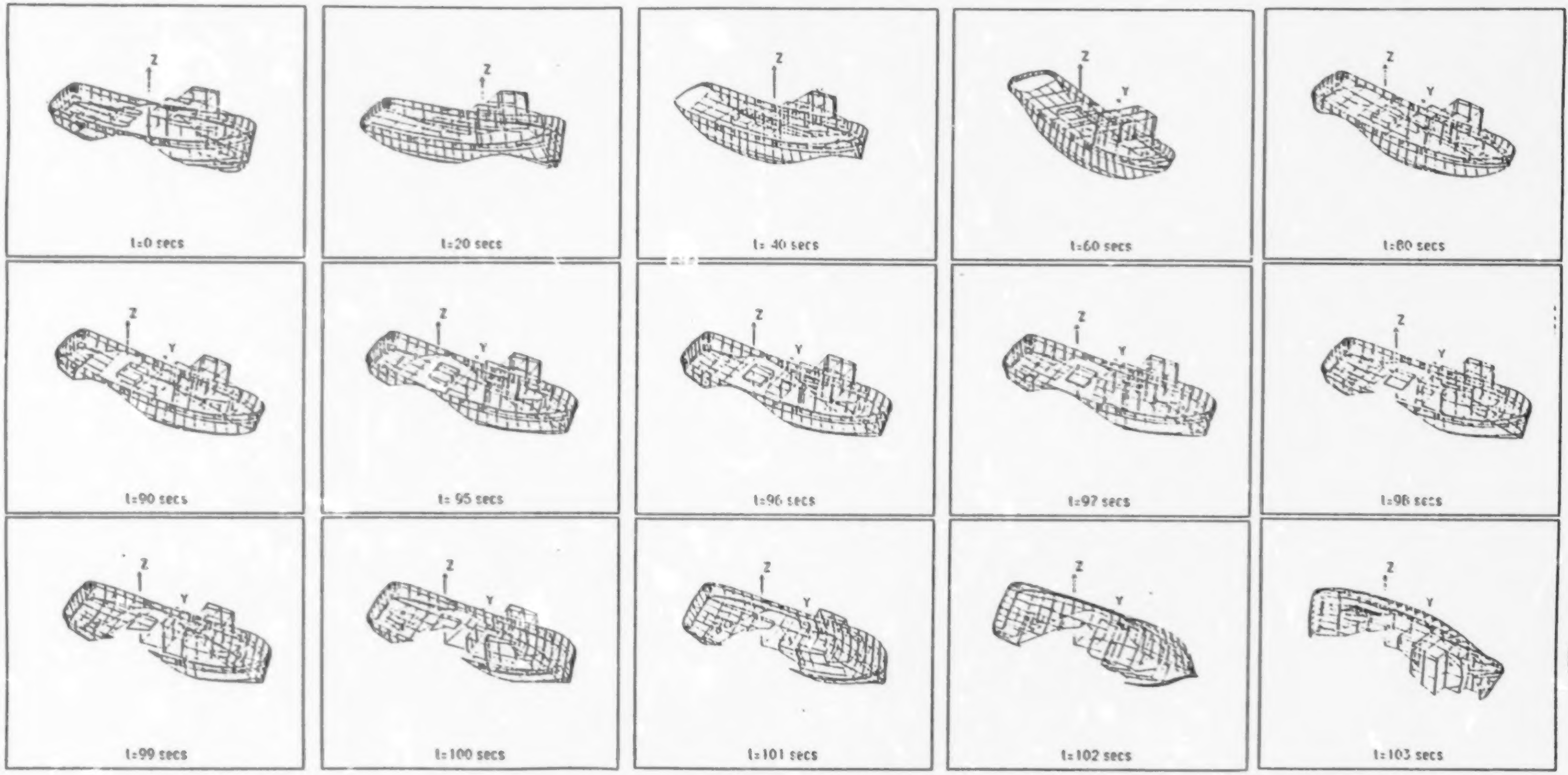
Roll

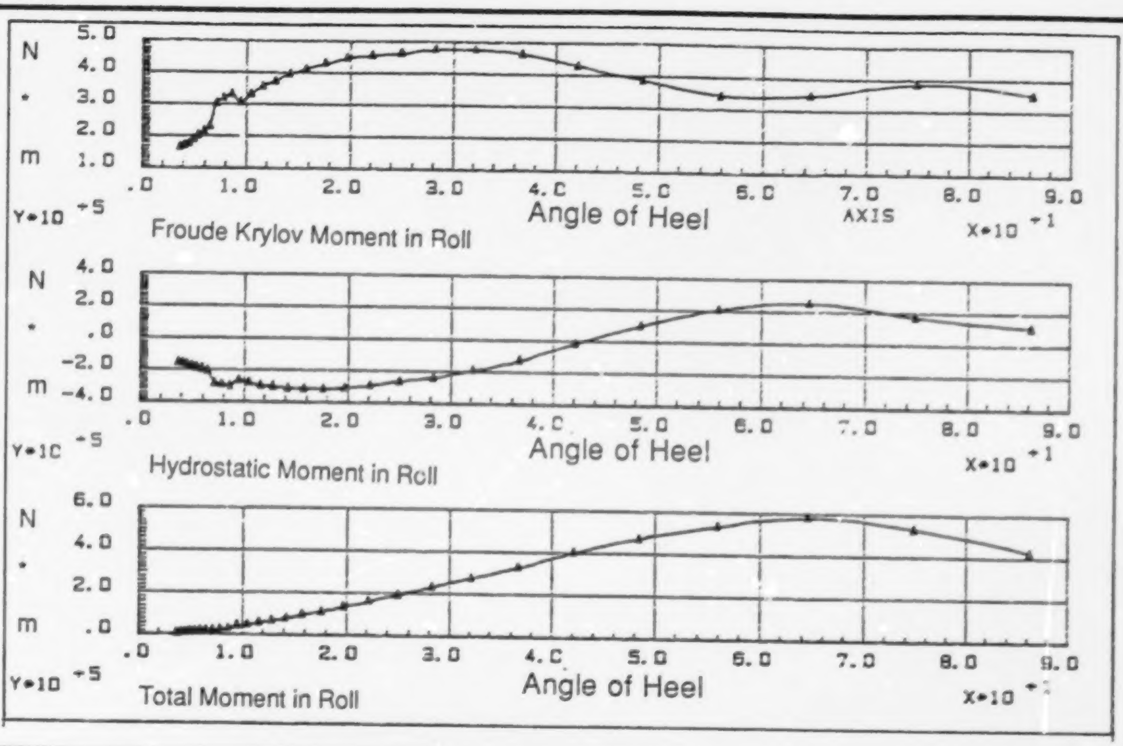
Pitch

Yaw

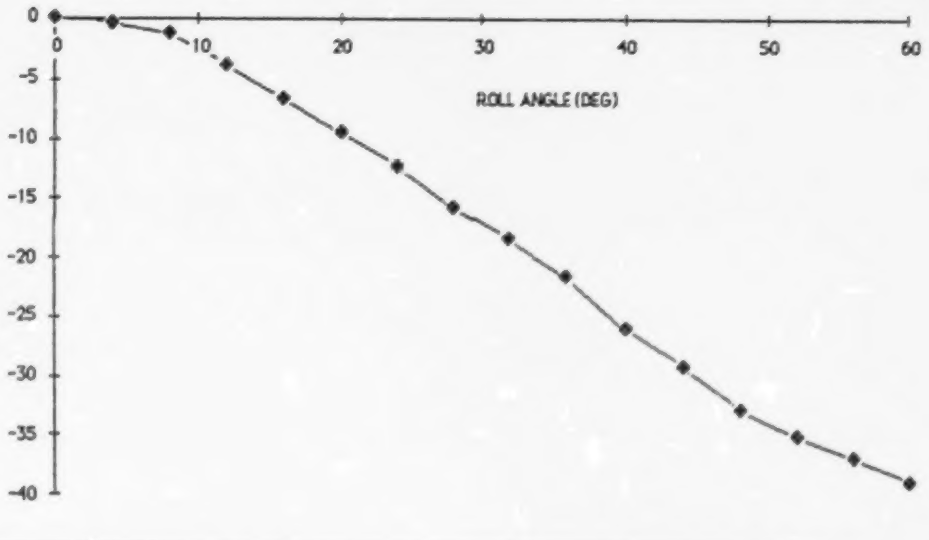
Figure 6.3 Motion History of "Hard Chine" Vessel in Case WLS4

Figure 6.4 Capsize Sequences of "Hard Chine" Vessel in Case WLS4


Z
Y
t=0 secs
t=20 secs
t= 40 secs
t=60 secs
t=80 secs
t=90 secs
t= 95 secs
t=96 secs
t=97 secs
t=98 secs
t=99 secs
t=100 secs
t=101 secs
t=102 secs
t=103 secs

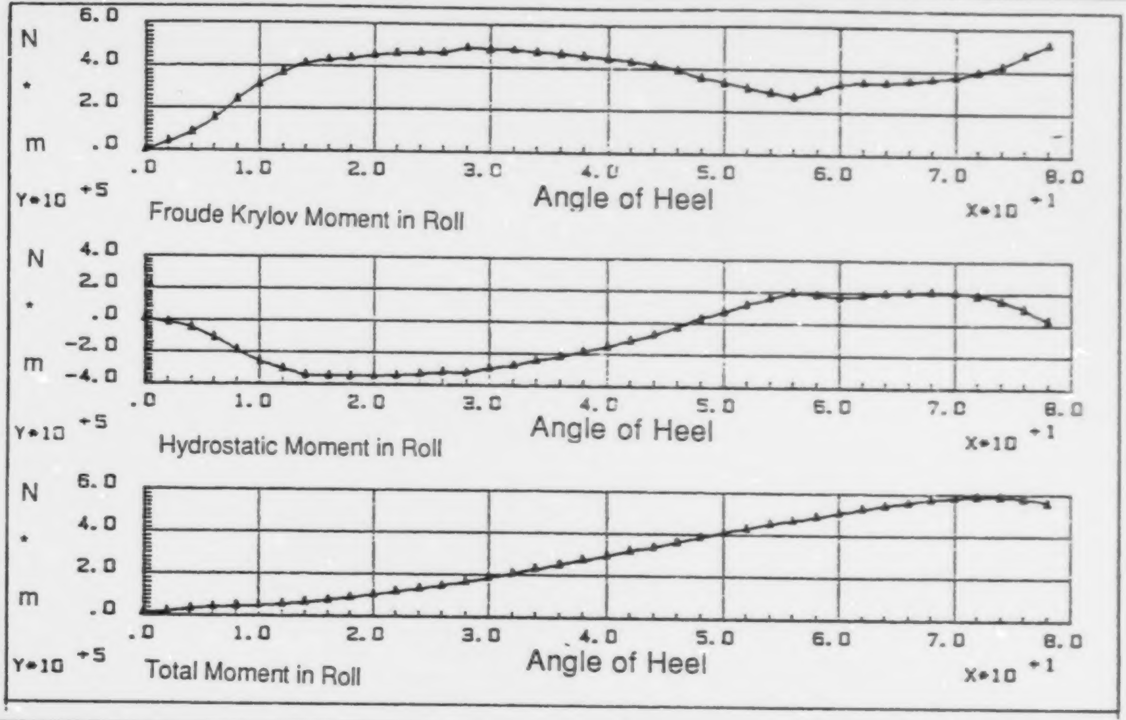


(a) Variations of roll moments

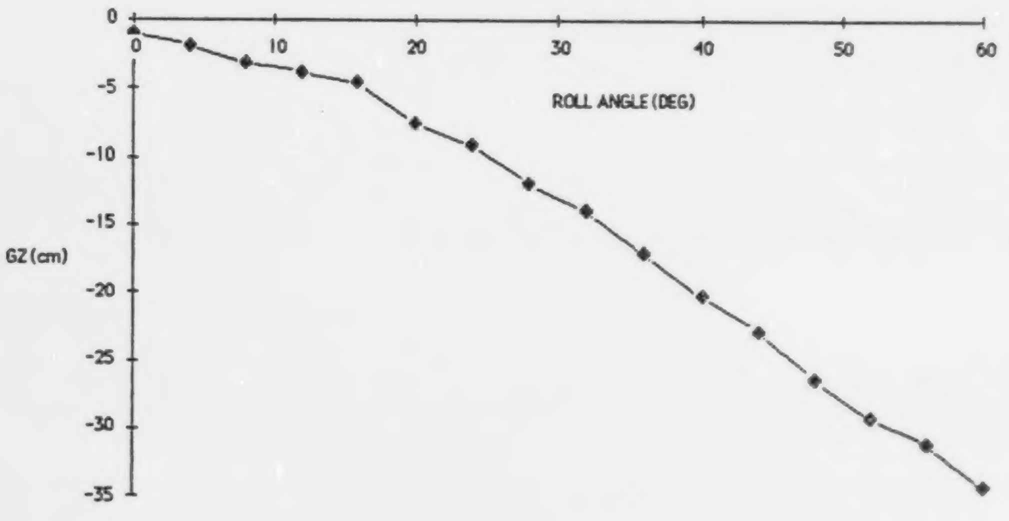


(b) Corresponding GZ curve

Figure 6.5 Dynamic Roll Moments and GZ Curve in a Wave Cycle for Case WLS4



(a) Righting moments



(b) Corresponding GZ curve

Figure 6.6 Righting Moments and GZ Curve with Wave Crest Amidships for Case WLS4

A further "quasi-static" inclining test was performed in the trough condition at 52.5 s . The results are shown in Figure 6.7(a). As the vessel is rolled to port the hydrostatic righting moment is greater than the Froude Krylov overturning moment, thus producing a net righting moment. The righting moment is totally diminished at angles exceeding 46 °. The corresponding GZ curve is given in Figure 6.7(b) (in open-diamond line) together with the curve of the crest amidships case (in solid-diamond line) for comparison.

A "quasi-static" run was performed on the wave crest at the smaller wave height (1/12 L) using an initial x-y-z position from WLS1 (at 99 s). The results of the test are shown in Figure. 6.8. It can be seen that a righting moment exists up to an angle of 20 °. This explains why the WLS1 case does not capsize.

6.6 Effect of Damping

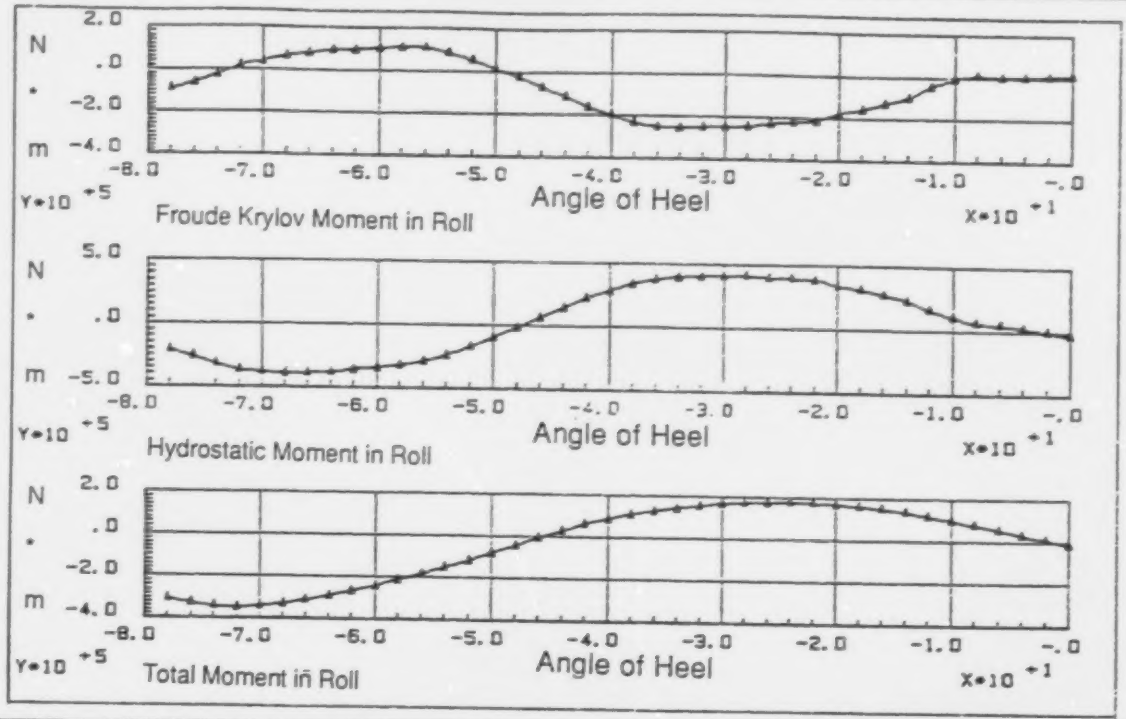
In this mode roll damping has little effect on the stability of the vessel since the roll motions of the vessel are small except at capsize. In order to confirm this case WLS4 (5 °, 1/7L) was repeated with roll damping of 10 % critical (Figure 6.9). By comparing the time histories in Figures 6.3 and 6.9, it can be seen that they are very similar and capsize occurs at the same time and in the same manner, proving the added damping has little effect.

6.7 Comparison Between Vessels

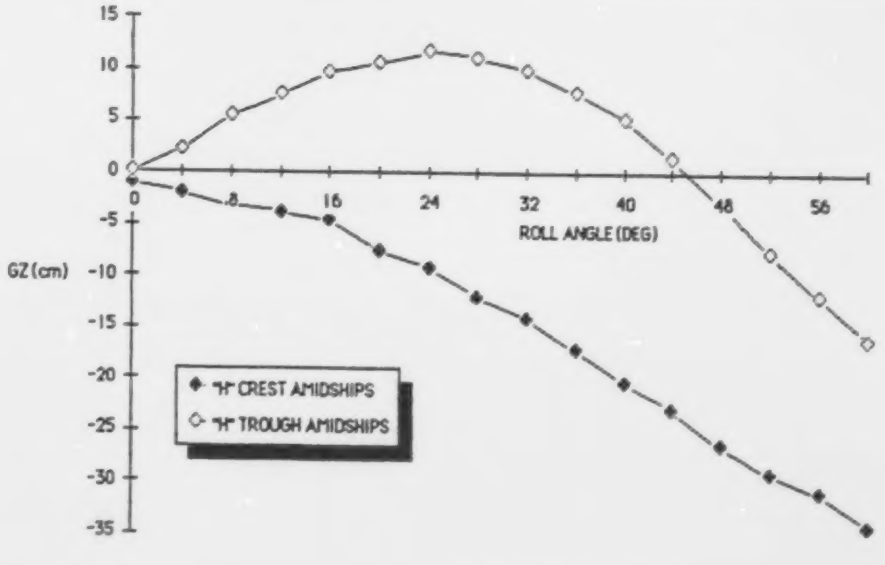
The "round bilge" vessel was run for all three wave directions at the higher wave height and capsize occurred in the 5 ° and 50 ° cases. This indicates that the "round bilge" vessel is no better in this mode of capsize than the "hard chine" vessel. As shown in previous section that damping has little effect in this mode of capsize, hence no additional runs were made for higher level of critical damping.

Figures 6.10 shows the motion time histories of case ELS4 for the "round bilge" vessel in the steep following 5 ° sea. They are essentially the same as the case WLS4 for the "hard chine" vessel. There is no rapid build-up of roll motion to a large amplitude prior to the capsize. The wave slowly overtakes the vessel, and at the same time it reduces the vessel's dynamic stability. The vessel then suddenly rolls over and capsizes at the next wave crest as the stability is completely lost. The enlarged plots of the Froude Krylov, hydrostatic and total moments in roll for the time period between 102.5 s to 108.5 s are shown in Figure 6.11. The "quasi-static" inclination run on the crest for case ELS4 is given in Figure 6.12.

Figure 6.13 compares the GZ curves of both vessels in the steep wave condition. As noted, both vessels lose stability to a greater or lesser degree when riding with the wave, and the worst condition occurs when the vessel is balanced on the crest of the wave.

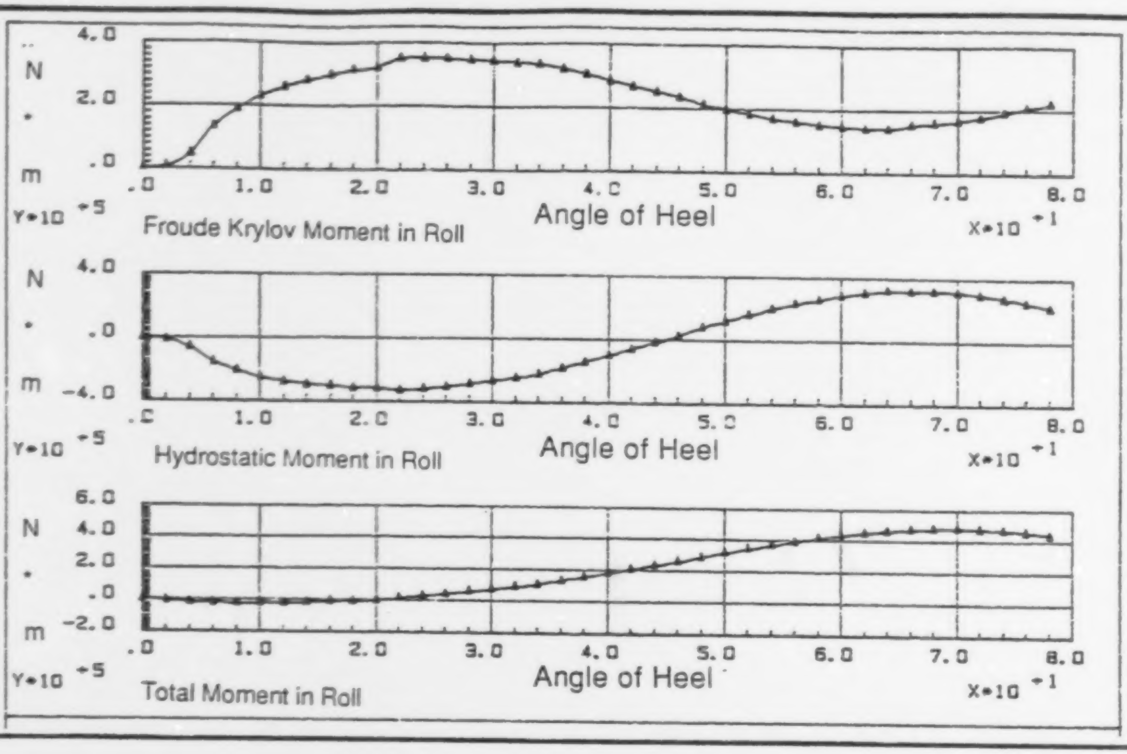


(a) Righting moments

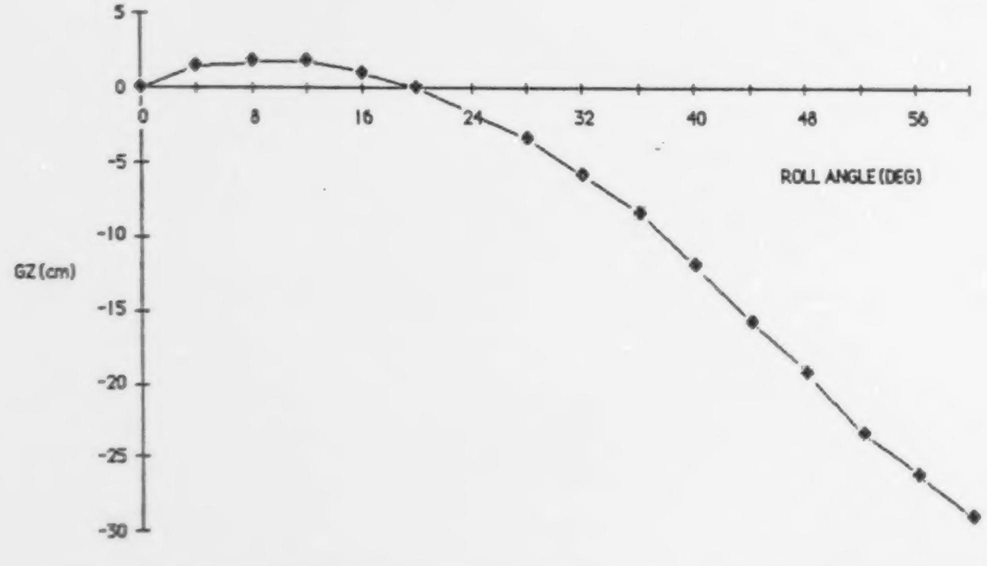


(b) Corresponding GZ curve

Figure 6.7 Righting Moments and GZ Curve with Wave Trough Amidships for Case WLS4

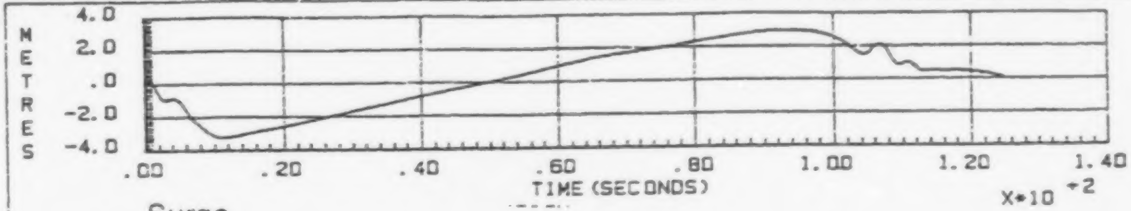


(a) Righting moments

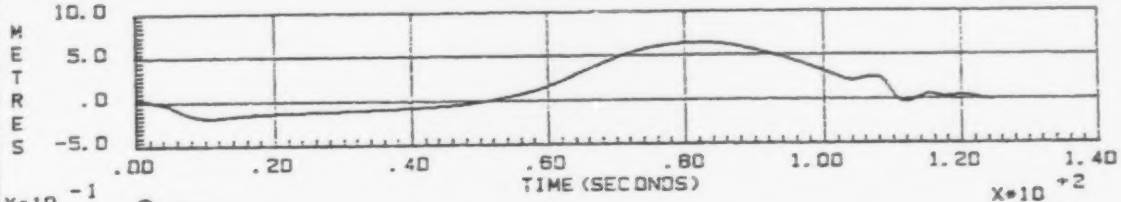


(b) Corresponding GZ curve

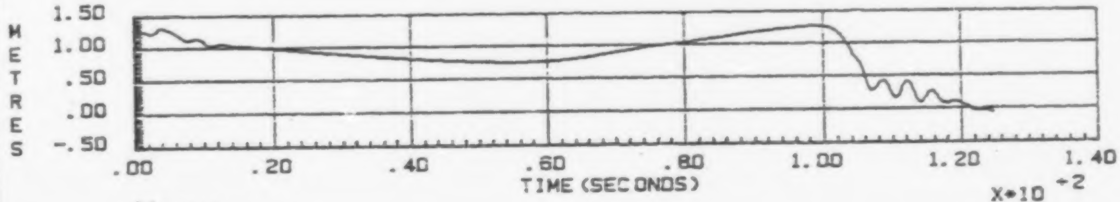
Figure 6.8 Righting Moments and GZ Curve with Wave Crese Amidships for Case WLS1



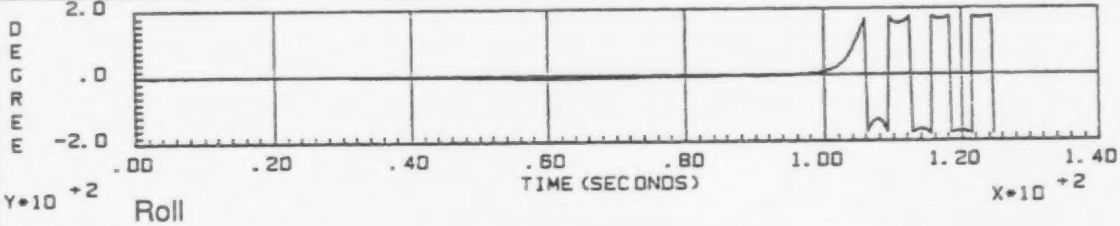
Surge



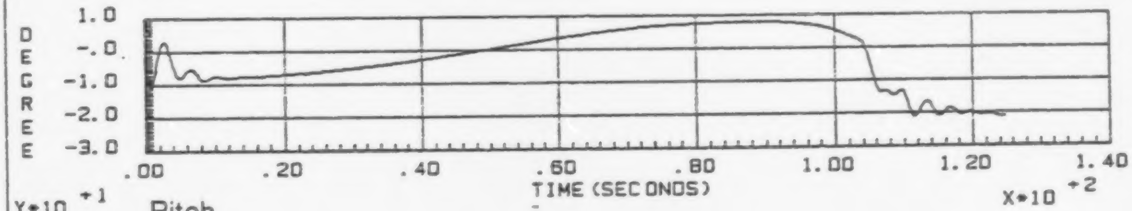
Sway



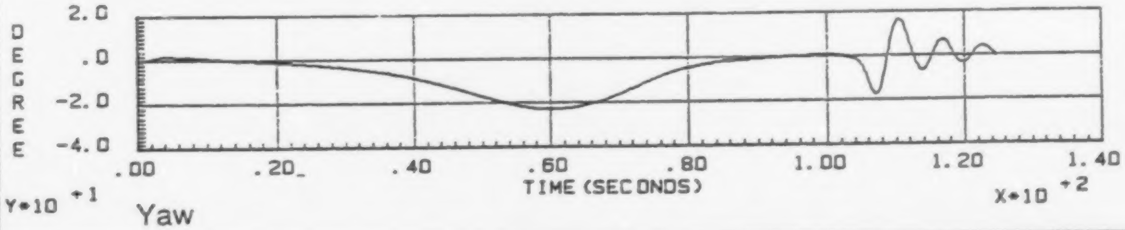
Heave



Roll



Pitch



Yaw

Figure 6.9 Motion History of "Hard Chine" Vessel in Case WLS4-10%

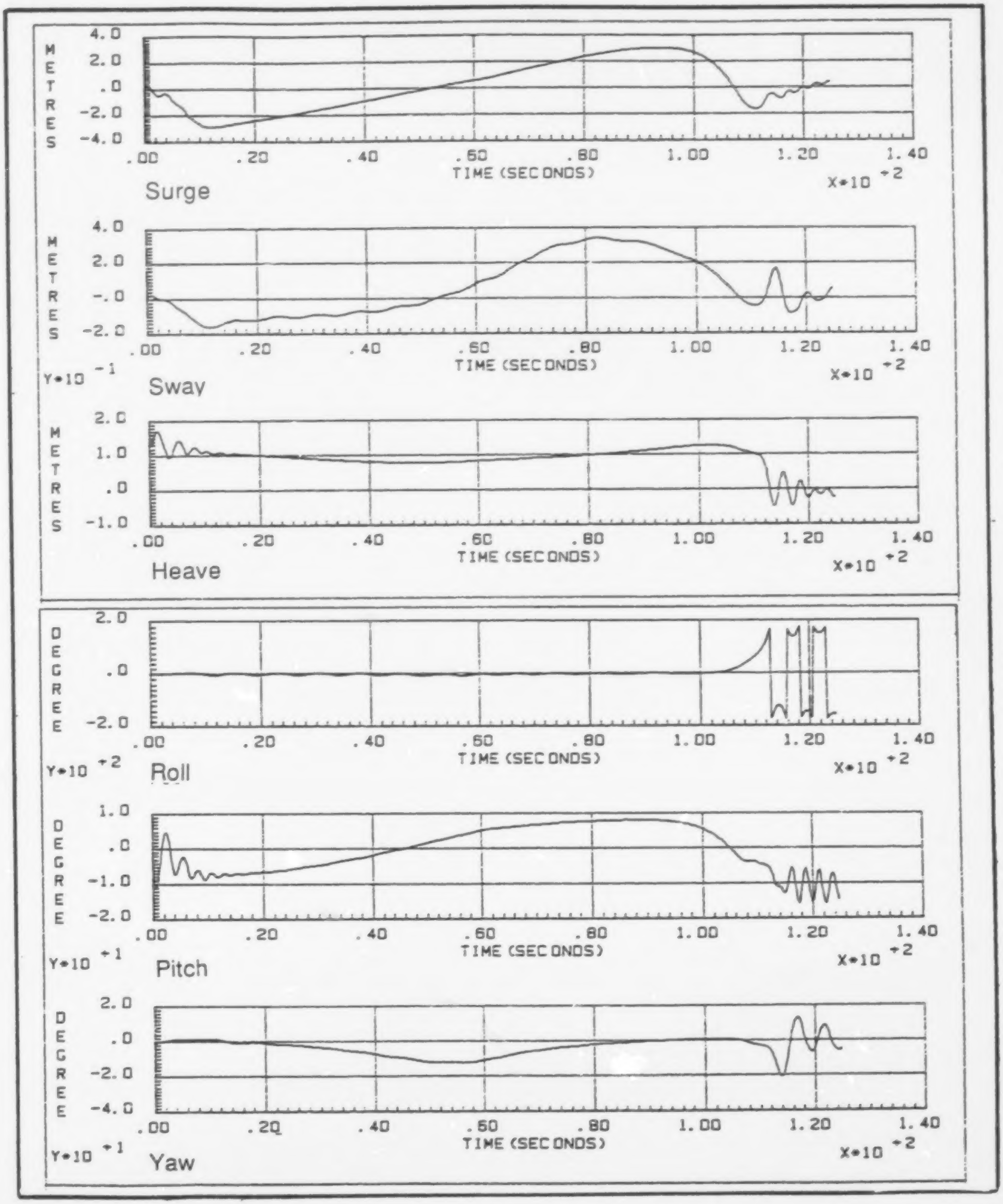
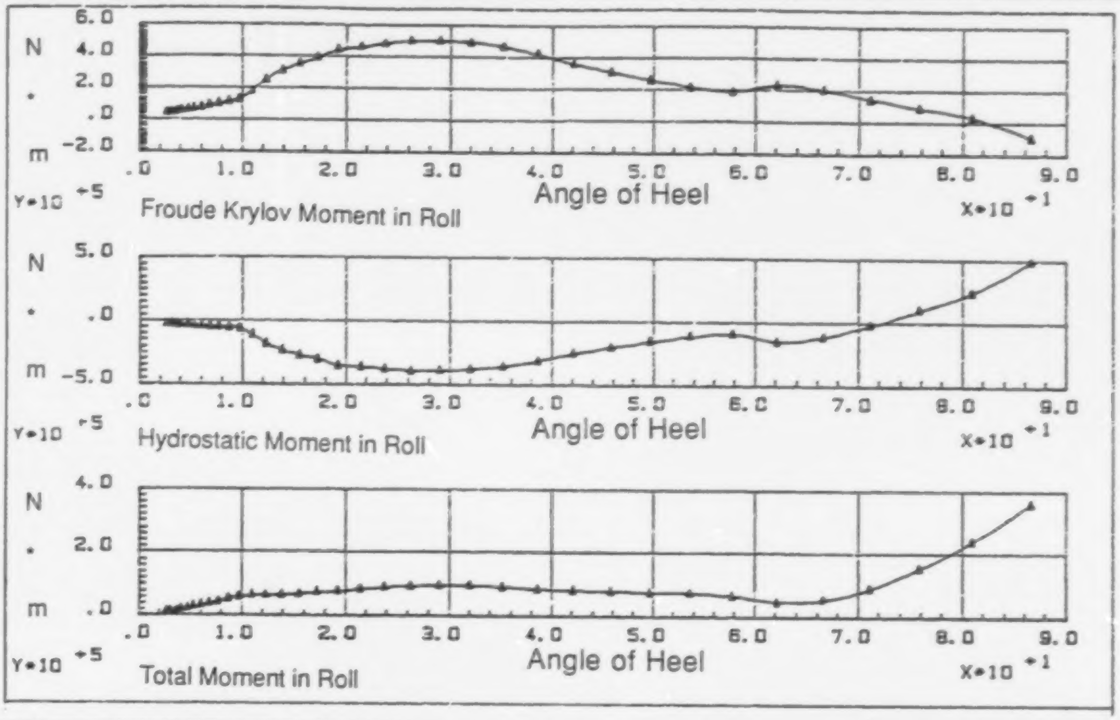
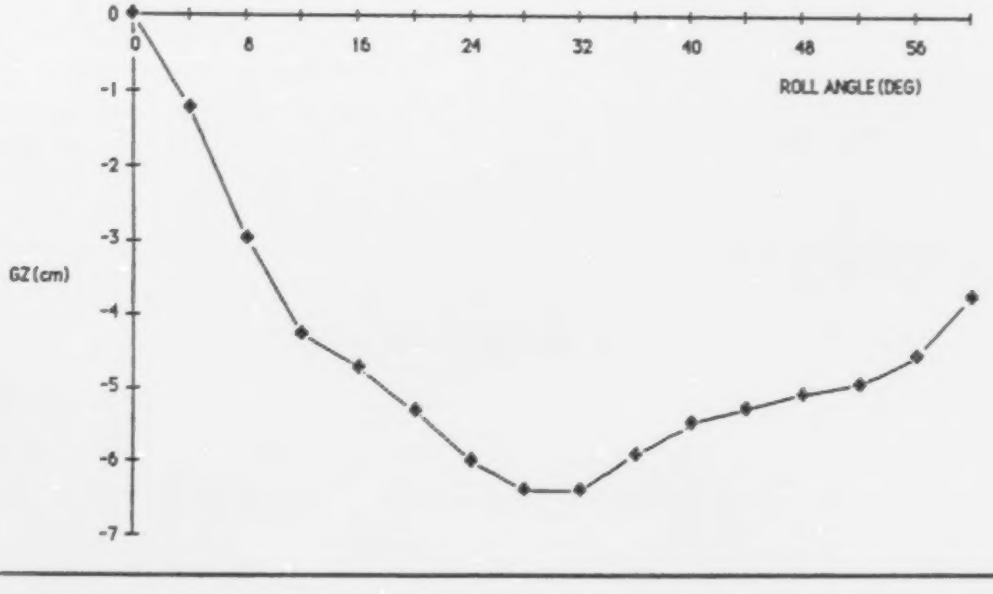


Figure 6.10 Motion History of "Round Bilge" Vessel in Case ELS4

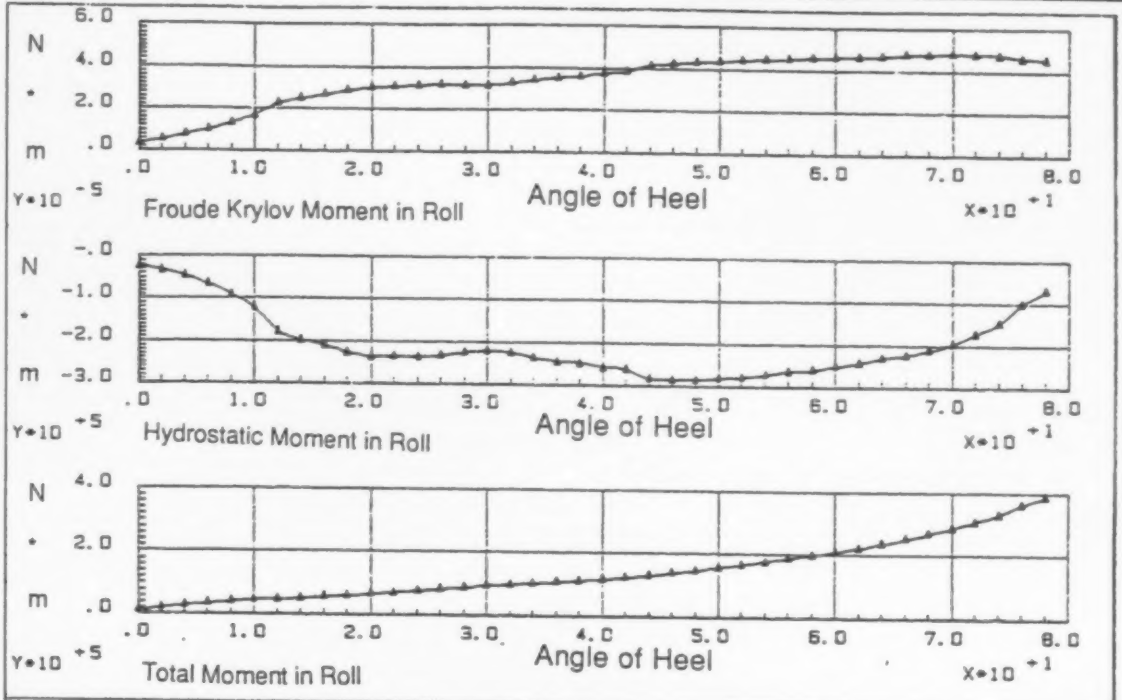


(a) Righting moments

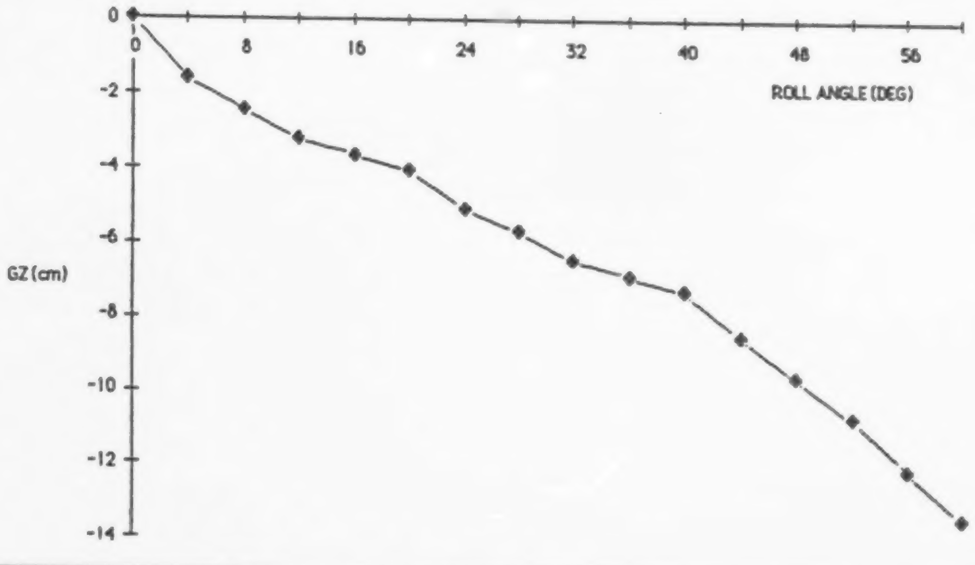


(b) Corresponding GZ curve

Figure 6.11 Dynamic Roll Moments and GZ Curve in a Wave Cycle for Case ELS4



(a) Righting moments



(b) Corresponding GZ curve

Figure 6.12 Righting Moments and GZ Curve with Wave Crest Amidships for Case ELS4

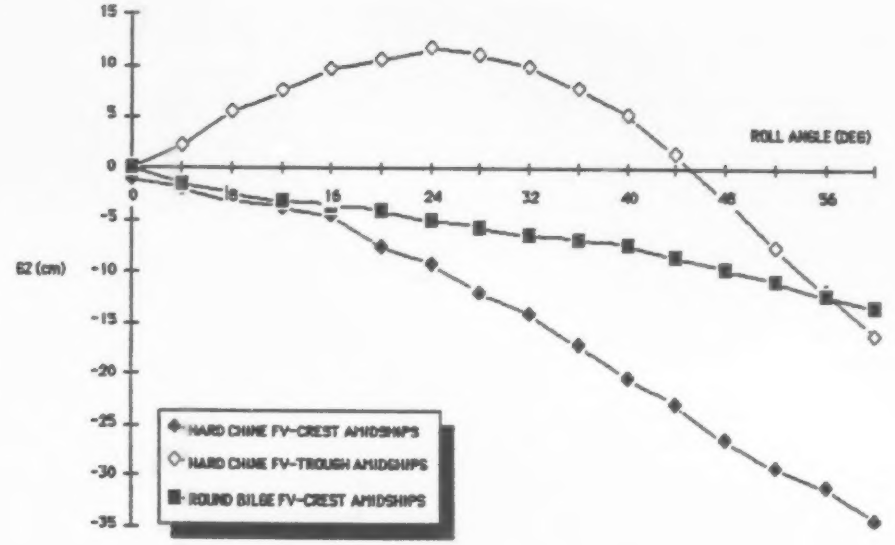


Figure 6.13 Comparison of GZ Curves Between Vessels in Waves for the Steep Following 5 ° Sea

For the near quartering sea conditions (following 30 ° and 50 ° waves), it is interesting to note the conflicting performance between the two vessels under the same wave condition; one vessel capsized while the other did not.

Figures 6.14 and 6.15 show the motion time histories for both vessels in the steep following 30 ° sea (WLS5 and ELS5), where the "hard chine" vessel capsized and the "round bilge" vessel did not. Both vessels suffered large reduction in stability during the passage of the first wave. As the second wave crest moved amidships, the "hard chine" vessel had completely lost its dynamic stability, while the "round bilge" vessel yet maintained a small, but positive instantaneous righting moment. The former can be seen in Figure 6.16 - at 101 s the crest is amidships and the positive total roll moment for a positive roll angle indicates a upsetting moment, thus zero stability. The latter is shown in Figure 6.17 - at 103.5 s the crest is amidships and the negative total roll moment for a positive heel angle (about 1 °) denotes a positive righting moment. The "quasi-static" GZ curves for both vessels with the crest amidships are given in Figure 6.18. For the "round bilge" vessel, a positive range of righting arms exists up to an angle of 22 °, but none at all for the "hard chine" vessel. This explains why the "hard chine" vessel capsized under this wave condition.

Shown in Figures 6.19 and 6.20 are the motion time histories for both vessels in the steep following 50 ° sea (WLS6 and ELS6). In this sea condition the "hard chine" vessel remained stable but the "round bilge" capsized. Figure 6.21 shows the enlarged view of the roll motion and moment time histories of the "hard chine" vessel for the time period from 100 to 116 s, as the second wave slowly moved to amidships. With the crest amidships at 108 s, the vessel maintained a positive righting moment (negative total roll moment for a positive roll angle). The dynamic stability continued to improve as the wave slowly passed amidships. On the contrary, there existed no positive righting moment for the "round bilge" vessel with the crest amidships at 106 s, as shown in Figure 6.22. Because of the positive instantaneous GZ at the wave crest the "hard chine" vessel was able to restore to its dynamic stable position and survived.

A close examination of the roll motion, velocity and acceleration time histories for the capsized cases (WLS5 and ELS6) as given in Figures 6.16 and 6.20, it is noted that these three components are all in phase as the second wave approaches amidships. The inertia moment (results of acceleration), which is normally 180 ° leading the roll motion, now works with the angular momentum to accelerate the heeling and carries it beyond the critical angle of zero righting arm as the wave passes amidships. On the other hand, the phase relationship between these three components is restored at the end of the passage of the first wave for the stable cases (WLS6 and ELS5). As can be seen in Figures 6.15 and 6.19, the velocity leads the motion by 90 ° and the acceleration leads the motion by 180 °.

The above performance comparison indicates that capsize from loss of stability during prolonged presence on a wave crest, while travelling at or near wave speed in a following sea, is a danger for both vessels examined. The difference in hull form appears to have little effect on their seakeeping performance.

Analysis results for the two stable cases in the near quartering sea conditions (WLS6 and ELS5) suggest that capsizing of the vessel in this model can possibly be prevented by maintaining a positive instantaneous GZ on the wave crest.

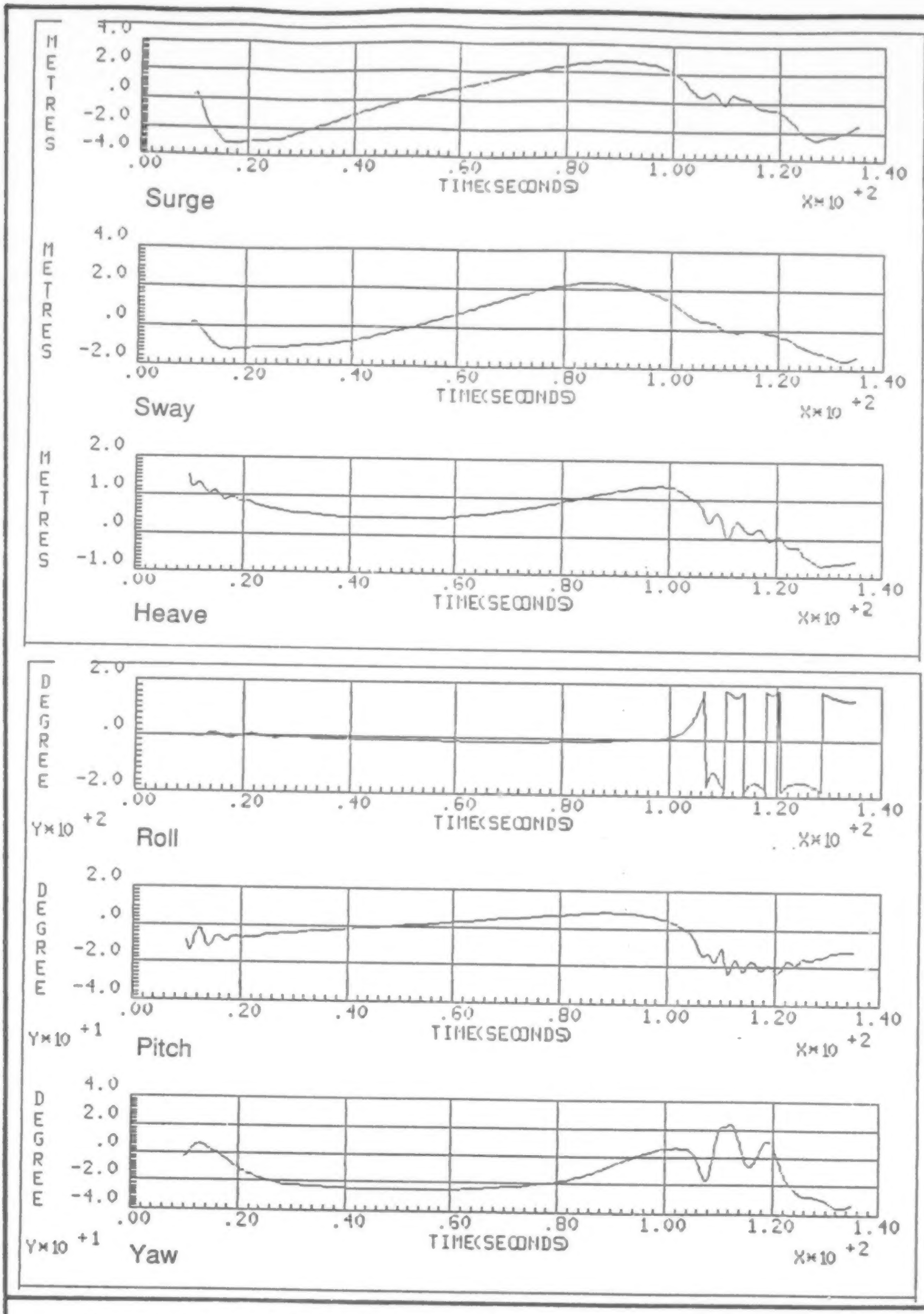


Figure 6.14 Motion History of "Hard Chine" Vessel in Case WLS5

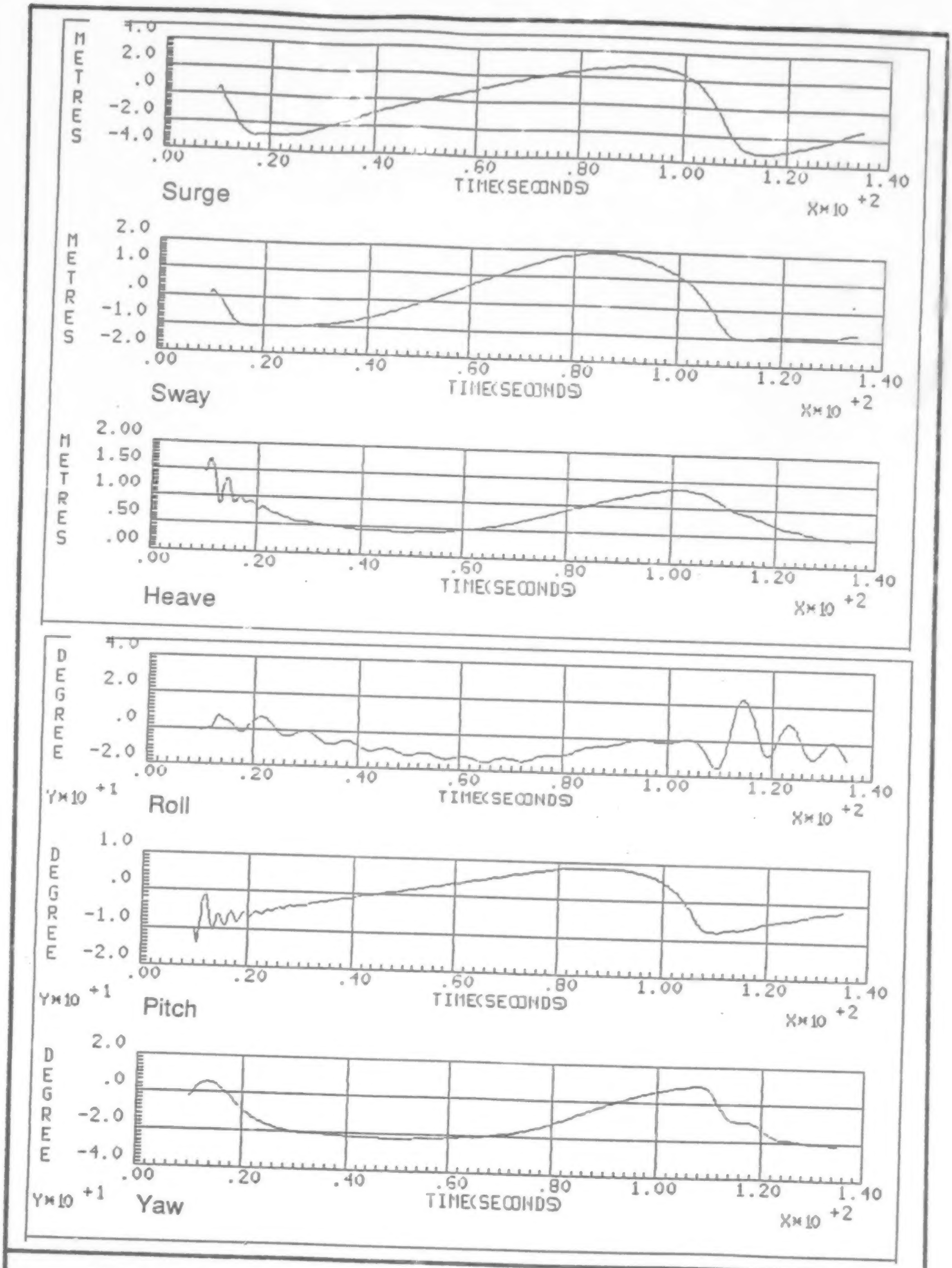


Figure 6.15 Motion History of "Round Bilge" Vessel in Case ELS5

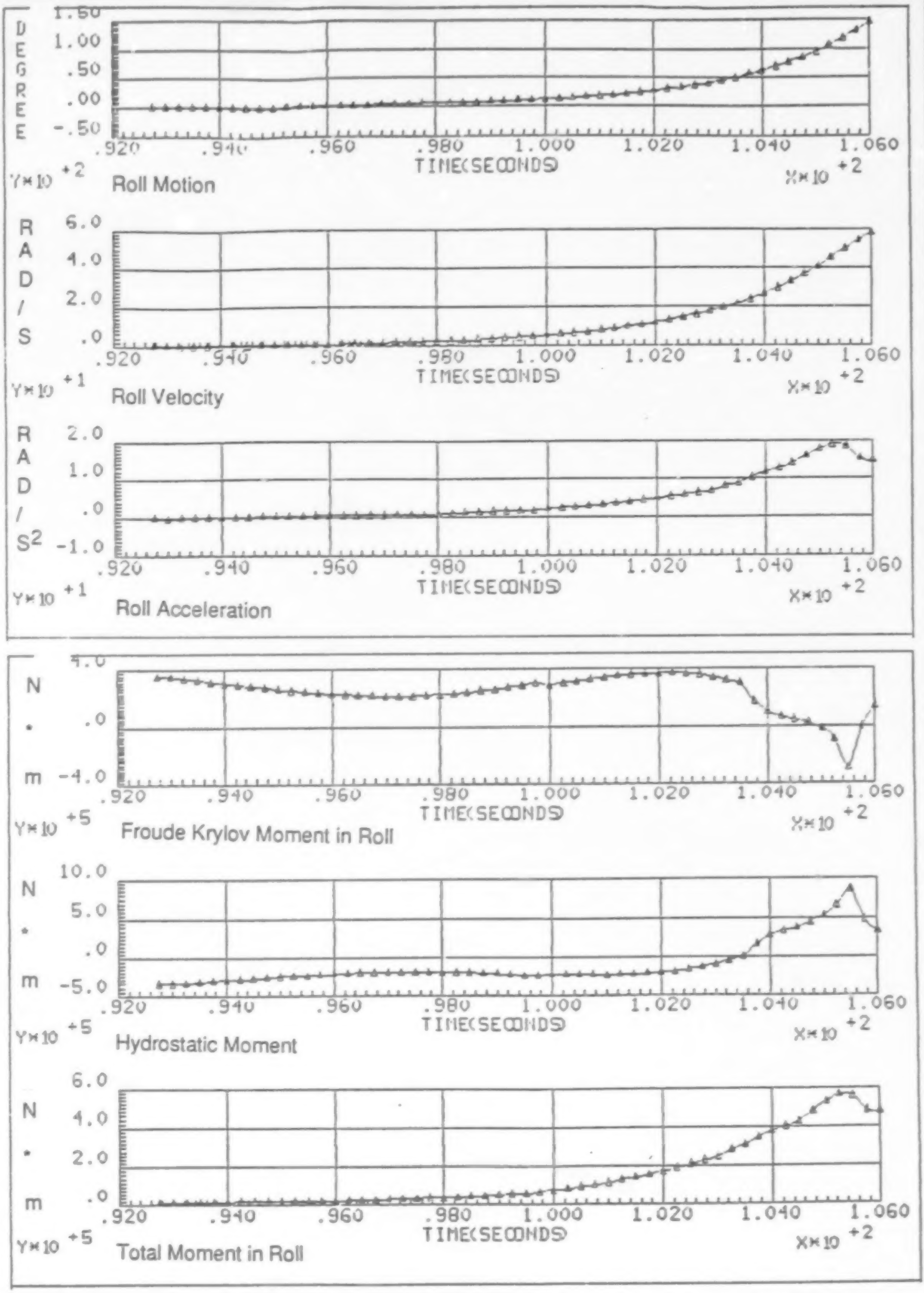
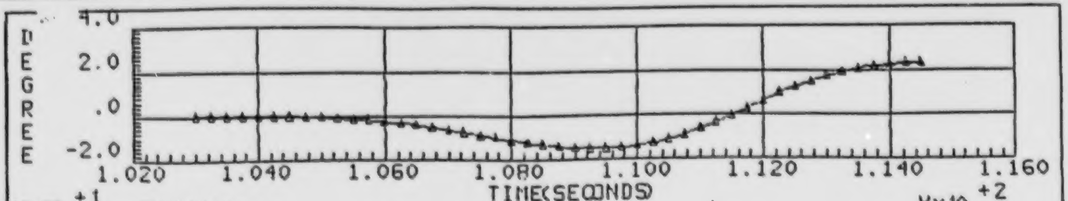
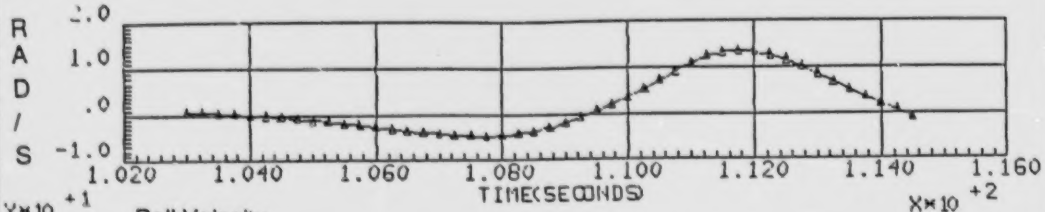


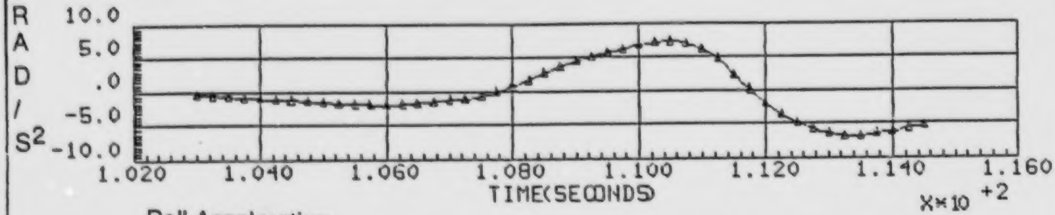
Figure 6.16 Roll Motion, Velocity and Acceleration and Roll Moments History Between 92 and 106 s for Case WLS5



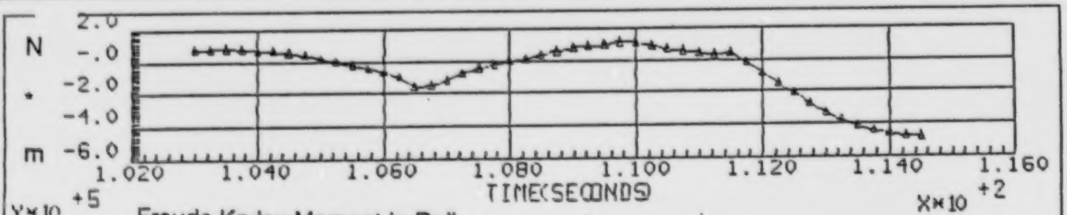
Roll Motion



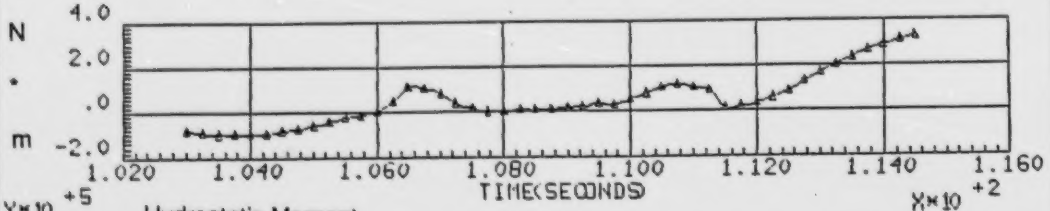
Roll Velocity



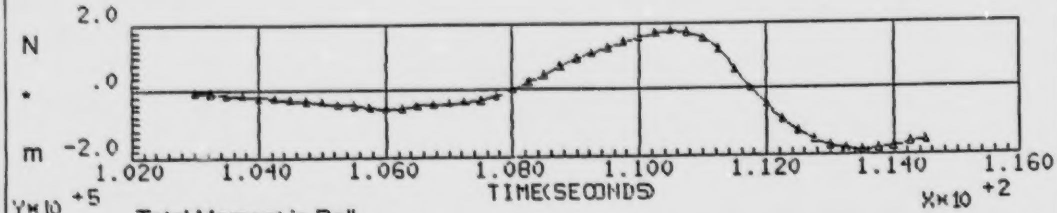
Roll Acceleration



Froude Krylov Moment in Roll



Hydrostatic Moment



Total Moment in Roll

Figure 6.17 Roll Motion, Velocity and Acceleration and Roll Moments History Between 103 and 115 s for Case ELS5

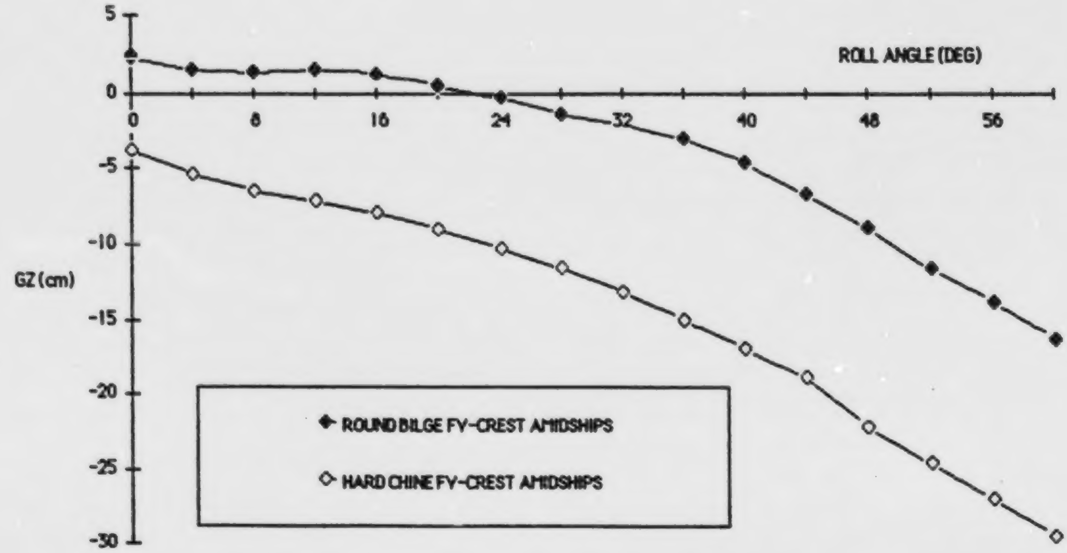


Figure 6.18 GZ Curves of Vessels on Wave Crest in a Steep Following 30 ° Sea

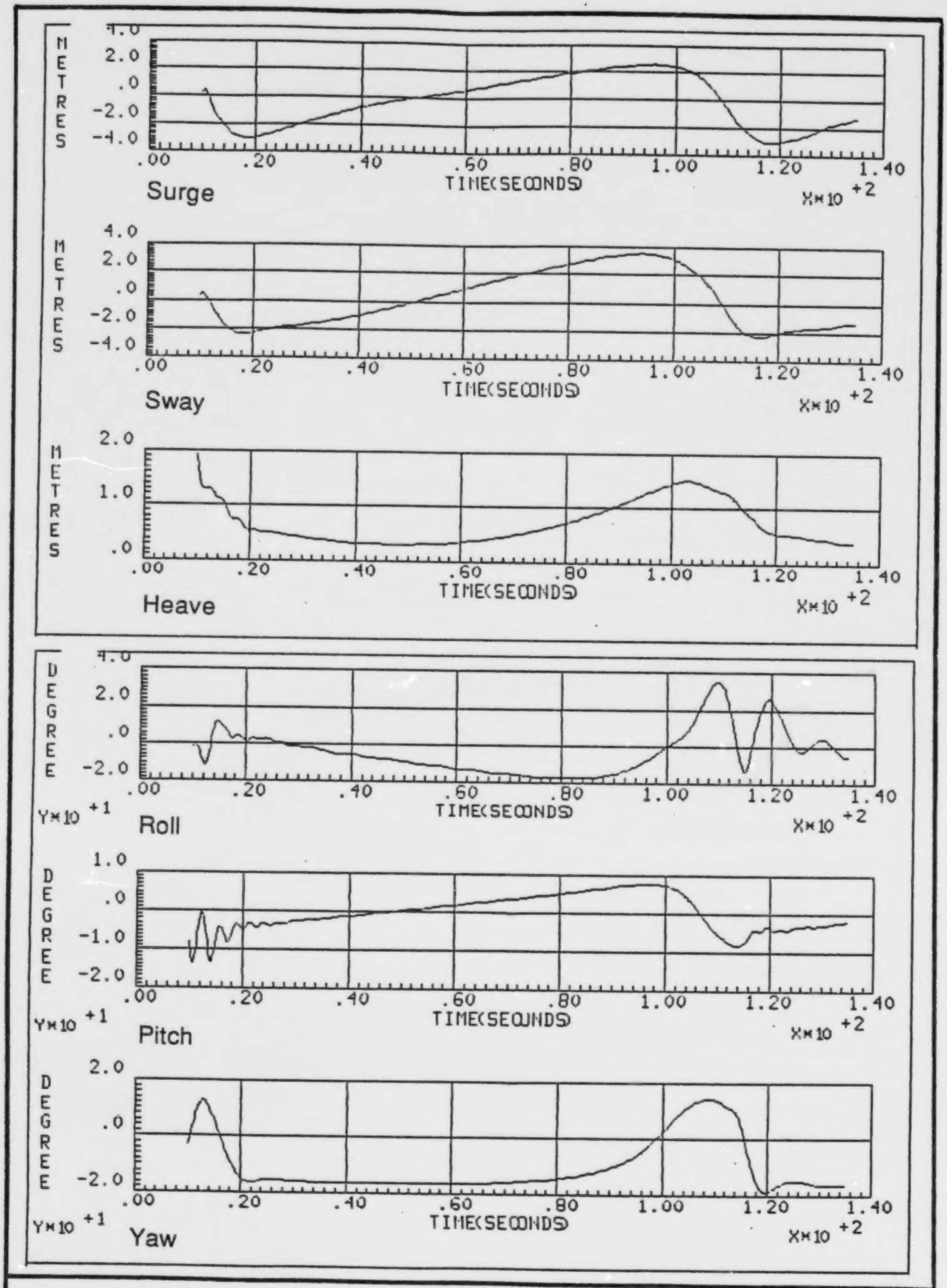


Figure 6.19 Motion History of "Hard Chine" Vessel in Case WLS6

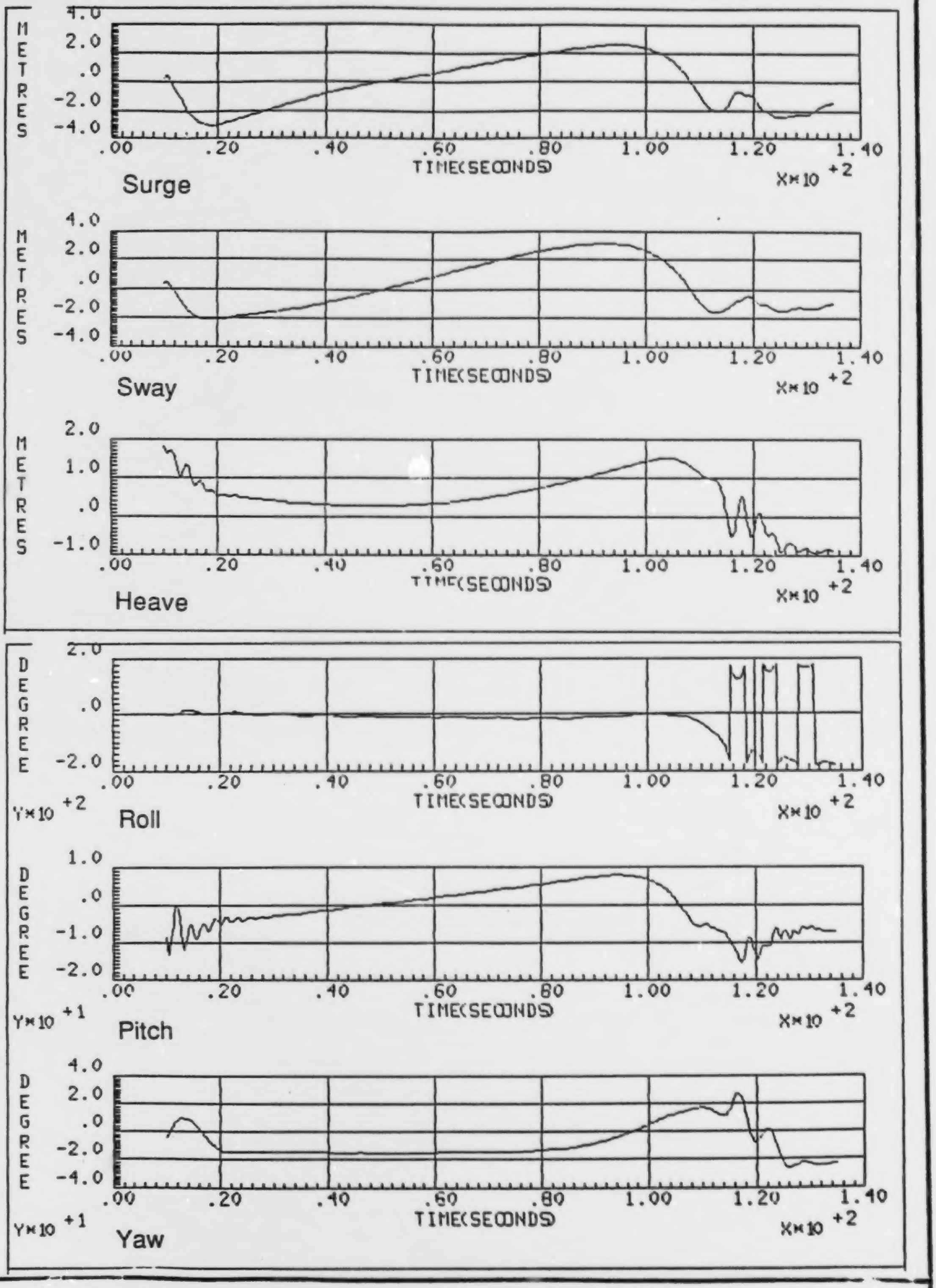


Figure 6.20 Motion History of "Round Bilge" Vessel in Case ELS6

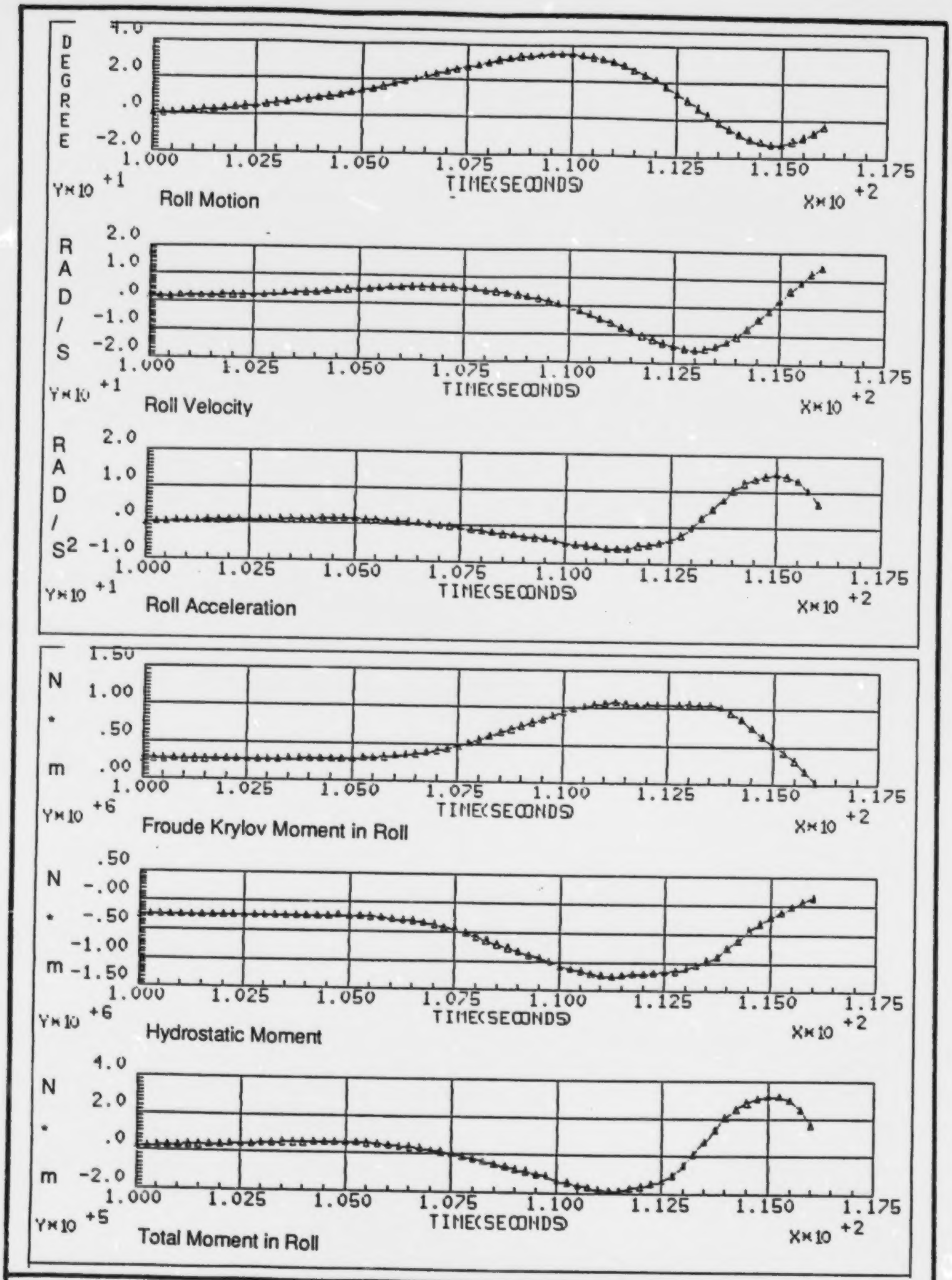
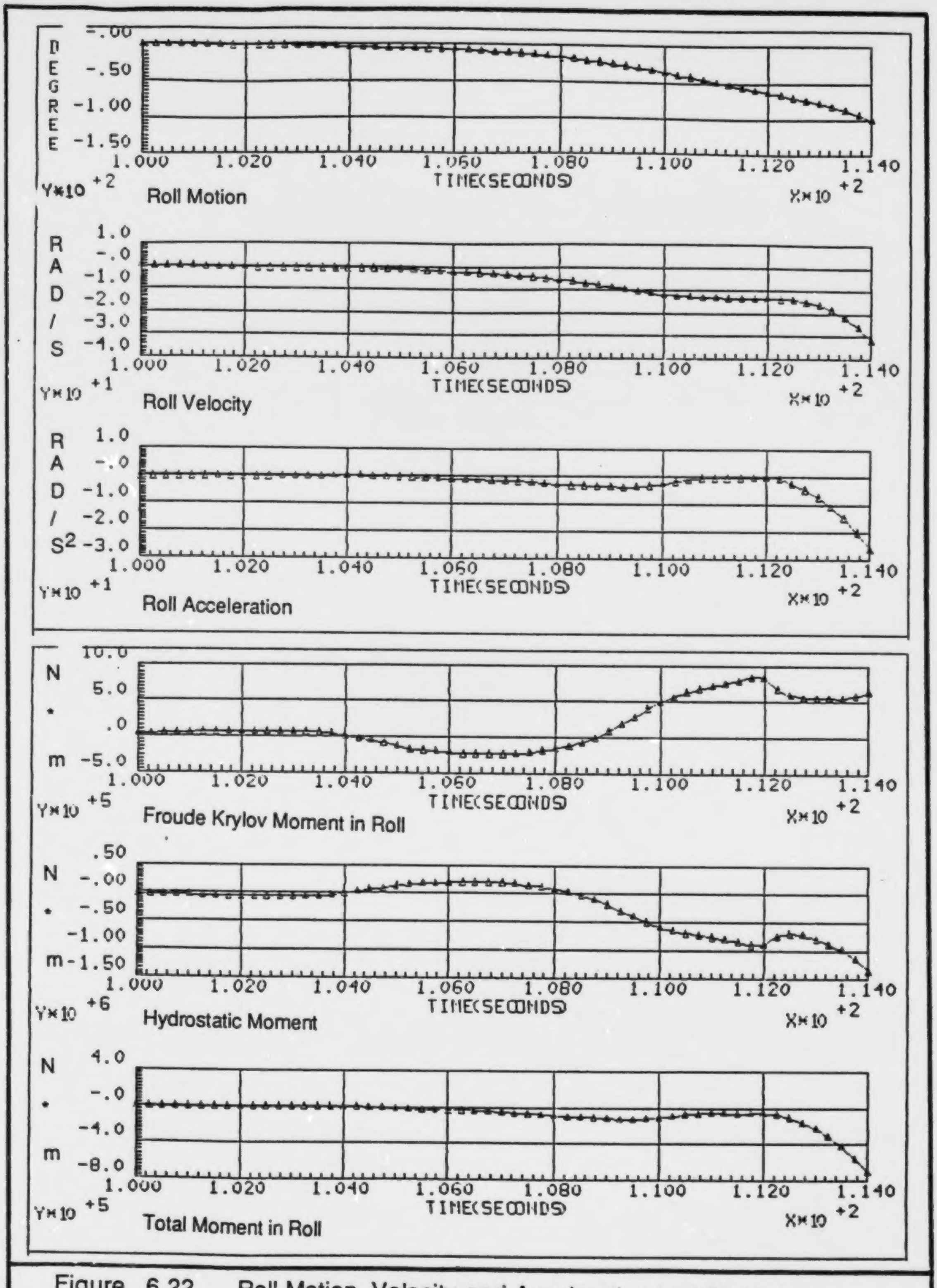


Figure 6.21 Roll Motion, Velocity and Acceleration and Roll Moments History Between 100 and 106 s for Case WLS6



Roll Motion

Roll Velocity

Roll Acceleration

Froude Krylov Moment in Roll

Hydrostatic Moment

Total Moment in Roll

Figure 6.22 Roll Motion, Velocity and Acceleration and Roll Moments History Between 10 and 114 s for Case ELS6

7. ROLL DAMPING MOMENT COEFFICIENT

The hydrodynamic damping forces acting on a ship during roll motion can be due to any combination of the following :

1. wave generation,
2. water friction on ship surface or eddy making,
3. hull form and appendages,
4. friction between ship and air,
5. energy loss because of heat generation during roll motion, and
6. surface tension.

The effects due to causes 1, 2, and 3 are significant, whereas those due to causes 4, 5, and 6 are considered to be very small.

In general, the roll damping is proportional to the square of the relative velocity between the marine vessel and its surrounding fluid, and it is also frequency dependant.

The roll damping as calculated by AQWA-LINE which was just 0.4 % critical was due to the wave making effect only. Additional damping due to effects of water friction and eddy making was added for the time domain simulations.

As shown, the capsize from subharmonic rolling requires light damping and the vessel performance in the prolonged presence on a wave crest is not influenced by damping, thus the values of roll damping assumed in these analyses are in no dispute. However, the results of the capsize from resonant rolling are noted to be sensitive to roll damping, thus the value of roll damping assumed in the computer simulation is subjected to some discussions.

It was noted that for conventional warships (long thin ships) the roll damping is in the range of 15 to 20 % critical at the ship's cruising speed and about 10 % at zero speed [2]. Morral [2] performed roll decrement experiments both on a trawler model and on a 57 m ARAB - Class trawler, and reported a roll damping ratio of 0.03 (3 % critical) at zero speed and a value of 0.07 (7 % critical) at the vessel's cruising speed of 10 knots.

Noting that there is no particular criteria for roll damping and no data available for the vessels under consideration, a value of 10 % critical (damping ratio=0.1) based on Morral's experiments can be justified by the following :

1) similar types of vessel
2) geometrically similar, even the hull forms (Morral's vessel is twice as big as the ones under investigation)
3) ship speed for simulation is approximately the same as test speed.

The readers are therefore reminded that the conclusions for the resonant capsize, thus drawn and as described in the next section, are based on the analysis results for a roll damping ratio of 0.1 (10 % critical) which is believed to be a representative level for the vessels under consideration.

It can be concluded that if the roll damping is to be increased to a higher level than the one presently assumed, it will certainly improve the seakeeping qualities of both vessels, and may reduce the probability of capsize for the "hard chine" vessel. However, the determination of a more accurate roll damping characteristics which is truly representative of the vessels is not an easy task and might be a subject for future study.

Based on the analysis findings on the effects of roll damping, it would seem appropriate that the present stability rules should consider a requirement of minimum roll damping. This has also been discussed in [2]. The requirement will improve safety, particularly for vessels which are lightly damped. The most widely used as well as the simplest type of device to increase damping is the bilge keel. The damping moment is generated by a component supplied by the pressure resistance of the bilge keel itself and by a component due to the change in the pressure distribution on the hull. Other means include gyroscopic stabilizers, tank stabilizers and stabilizing fins. In respect to the criteria for minimum roll damping, this study and data from other studies are insufficient to formulate such requirements. Further studies on damping requirements should be considered.

8. CONCLUSION

1. The study has shown that **hydrodynamic programs** such as those included in the AQWA suite utilized for this project, can be of value to regulatory agencies in investigating the dynamic stability of vessels in both the analysis of incidents and in the evaluation of proposed regulatory standards. Programs of this type can provide assistance in determining the hull form and stability characteristics to be avoided in order to reduce the risk of capsize. However, the size, complexity, availability and cost of such programs preclude the imposition of such programs in regulation.

2. The simulation undertaken, which assumed a **subharmonic rolling** in a wave encounter period of half the vessels natural roll period, confirmed that because very regular waves of precise period are required, roll motions only grow to dangerous levels if unrealistically low levels of damping are assumed. However, the results demonstrate that ensuring a large angle of vanishing stability is an effective measure to reduce the possibility of capsizing in this sea condition. In the steep following wave condition at the subharmonic periods, the "round bilge" vessel, which possesses a higher angle of vanishing stability, was shown to right itself, despite large roll motions up to 55 °.

3. In the case of **resonant rolling** in a wave encounter equal to the vessels natural period it was shown that the "hard chine" vessel would capsize in the steep waves at both levels of critical damping assumed. However tendency to capsize was reduced at higher critical roll damping factors.

   The "round bilge" vessel modelled was shown to have greater dynamic stability in waves than the "hard chine" vessel, especially on the wave crest, despite both vessels having the same GM. At angles below the angle of deck immersion, (about 30 ° for both vessels investigated) the under-water hull form dominates the curves of the righting arms (GZ) . Refer to Figure 5.15c . This results from differences in hull designs.

   Measures to reduce the danger of capsizing in this mode would include ensuring a positive small angle transverse stability in waves and to provide good damping characteristics (e.g. such as bilge keels). However the GZ curve for all angles of heel is of significance.

4. The simulation of the vessels **travelling at wave speed**, resulting in a prolonged presence on the wave crest, showed that both vessels lost stability and capsized.

5. In general, **variations in stability in waves** play an important role in the safety of small vessels, as illustrated in the last two modes of capsize examined. The vessels could suffer severe reduction of stability in certain positions relative to the wave, resulting in extreme motions or even capsize. The worst wave position is with the crest at or near amidships.

   The hull form aft plays a significant role on stability in certain positions relative to the wave profile. With the hull balanced on the crest, and with the stern emerging from the wave surface, the vessels lose waterplane inertia and hence stability.

6. The **effect of damping** can be important on the vessel motions. Increase in damping reduces the roll motions and can in turn produce greater dynamic stability and reduce the probability of capsizing. However, in the capsize cases of the vessels riding on top of, and with the wave crest, damping has little effect.

9. RECOMMENDATIONS

1. Within the framework of the present static stability rules, a modified regulation should be considered which would improve the safety of small fishing vessels by requiring a large angle of vanishing stability, i.e. positive GZ values up to 80 - 90 °, recognizing that small vessels will heel to much larger angles more often than large vessels.

2. To improve the safety of small fishing vessels considerations should be given to :

   (a) A requirement for a range of positive GZ at all vessel positions in waves.
   This is drawn based on the conclusions 3 and 5 as well as the analysis findings in section 6.7.

   (b) A minimum roll damping requirement.
   This is based on the analysis results on damping effect and the discussions in section 7.

In recognition of the limited data available from this and other studies, it may not be practical at present to formulate these regulations. Further case studies may be justified to expand the analysis with the objective of developing simplified assessment criteria for use in stability regulation. It is recommended that such studies be initiated to provide a technical basis to assist in the process of modifying regulations.

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APPENDIX A DEFINITIONS OF HYDROSTATIC AND FROUDE-KRYLOV FORCES WITHIN THE AQWA SUITE

The hydrostatic forces are calculated from the integral of the hydrostatic pressure over the wetted surface of the body. The hydrostatic moments are considered to be taken about the center of gravity of the body in question. The expression for the hydrostatic force ($\mathbf{F}_{hys}$) and moment ($\mathbf{M}_{hys}$) are as follows :

$$\mathbf{F}_{hys} = - \int_A \mathbf{P}\,\mathbf{N}\ dA$$

$$\mathbf{M}_{hys} = - \int_A \mathbf{P}\ (\mathbf{r} \times \mathbf{N})\ dA$$

where $\mathbf{P}$ = hydrostatic pressure (i.e. $- \rho g z$)
= position vector w.r.t. center of gravity
$\mathbf{N}$ = outward normal vector of the body surface
A = wetted surface of the body

The "Froude-Krylov" force and moments are calculated directly from the integral of the wave pressure acting over the wetted surface of the body. The moments are again taken about the center of gravity of the body. The force on each element is given by:

$$\mathbf{F} = - \mathbf{N} \int_A p(x,y,z)\ dA$$

where $p(x,y,z)$ is now the dynamic wave pressure (i.e. $- \rho\,[\,\delta \phi / \delta t\,] - 0.5\,\rho\,\nabla\phi$) which depends on the wave theory used.

For a second order, deep water wave theory, the dynamic wave pressure is given as:

$$p(x,y,z) = \rho g\ a\ (e^{kz} \cos(kx - \omega t) - 0.5\ k a e^{kz})$$

where $x = x \cos(\alpha) + y \sin(\alpha)$
k = wave number
ω = wave frequency
a = wave amplitude
α = wave direction
ρ = density of water

APPENDIX B CORRECTIVE MOMENTS IN YAW

The validity of the servo designs for countering broaching was examined in part by comparing the corrective moment in yaw against the excitation moment. The latter is a direct output from the program run, while the former can be obtained from taking the yaw angles times the servo design stiffness in yaw.

The yaw stiffness is given as $K_{66} = (I_{66} + A_{66}) * (2 \pi / T_n)^2$ where I_{66} is the mass moment of inertia in yaw, A_{66} is the added mass moment of inertia in yaw, and T_n is the natural period of the control system. In the case of the servo design 2, the natural period is 20 s, thus the yaw stiffness is 2,390 kN-m/rad.

Calculation of the corrective yaw moments were made for both vessels in the steep following 30 ° wave condition, cases WSM5 and ESM5-10%. The results for these two cases were plotted (in dotted line) in Figures B.1 and B.2 together with the total excitation moments (in solid line). In each of these figures the yaw motion is also given for reference. The comparison between the corrective moments and the excitation moments in the figures clearly indicate that the servo design used were perfectly satisfactory.

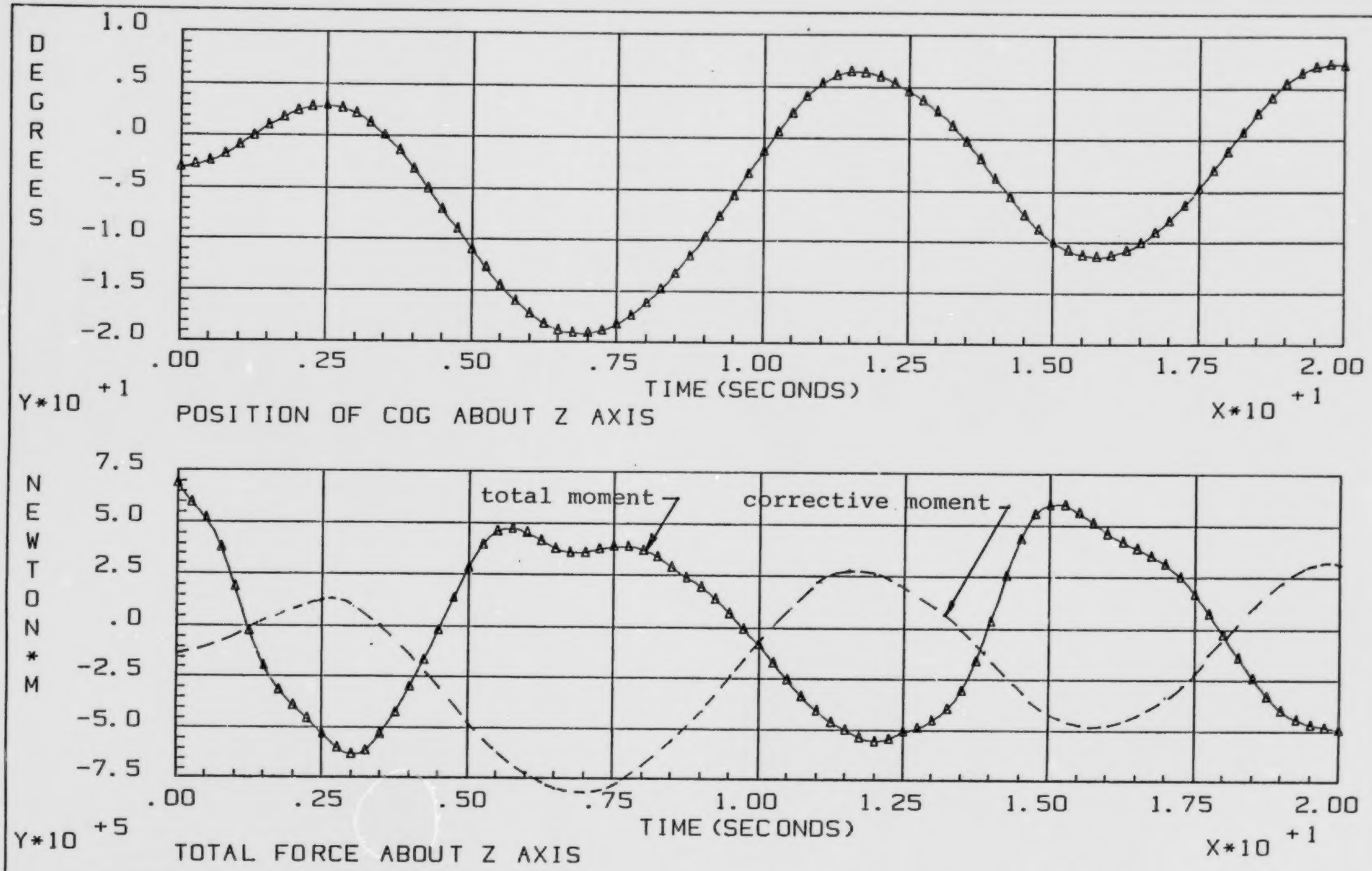

Figure B.1 **Comparsion Between Corrective Moment and Total Excitation Moment for Case WSM5**

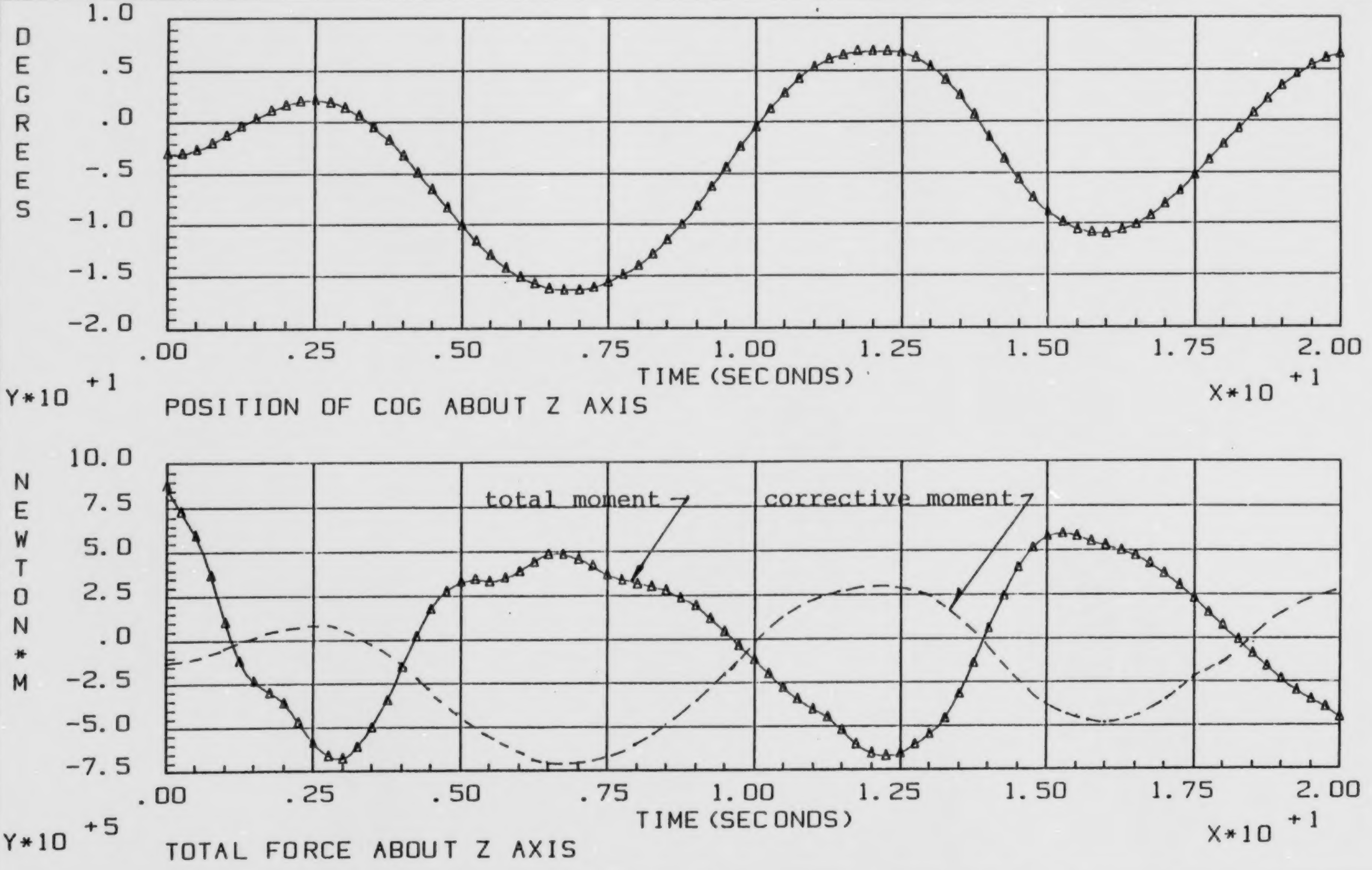

Figure B.2 Comparsion Between Corrective Moment and Total Excitation Moment for Case ESM5-10%

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TP 7407E

Investigation Of The Use Of Modern Ship Motion Prediction Models In Identifying Ships With a Larger Than Acceptable Risk Of Dynamic Capsizing

Volume 2
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F RESULTS OF " WAVE CREST " CASE

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N.R. Gore

17. Abstract

This report presents the results of the application of a suite of marine vehicle motion analysis computer programs to simulate the stability characteristics of fishing vessels. The programs utilized were developed by Atkins R&D (U.K.) for the use of the off-shore energy resources industry.

The performance of two 150-ton fishing vessels, one of a hard chine design and the other of a round bilge design, were simulated to compare the capsize behaviours of the two vessels in various sea conditions in an attempt to identify the influence of the different underwater hull forms.

The sea conditions and vessel speeds chosen for the computer simulations covered three known dangerous capsize mechanisms, vis: subharmonic rolling, resonant rolling and stability loss on a wave crest.

The results indicated that the round bilge vessel was somewhat more stable under the first two conditions, but both vessels capsized during prolonged presence in a wave crest while travelling at wave speed.

It was concluded that programs of the type used could be developed into useful tools to investigate the dynamic stability of vessels.

18. Key Words
Ship stability, fishing vessels, capsize mechanisms, computer simulations

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14. Code de l'organisme parrain: 6363/64/65

15. Remarques additionnelles

16. Agent de projet — CDT: N.R. Gore

17. Résumé

Ce rapport présente les résultats d'une série de simulations par ordinateur concernant les caractéristiques de stabilité des bateaux de pêche. Les programmes d'analyse utilisés avaient été élaborés par Atkins R & D (R.-U.) à l'intention de l'industrie des ressources énergétiques au large des côtes.

Les simulations ont porté sur deux bateaux de pêche de 150 tonneaux, différenciés par la forme de leur coque: coque à bouchains vifs dans un cas et coque à bouchains ronds dans l'autre. En comparant les caractéristiques de chavirement par différents états de mer, les chercheurs espéraient déterminer l'influence que pouvaient avoir différentes formes de carène.

Les états de mer et les vitesses d'évolution retenus matérialisaient trois conditions dangereuses connues pour être propices au chavirement, soit le roulis en mode sous-harmonique, le roulis en mode résonnant et la perte de stabilité sur crête de houle.

Les résultats montrent que le bateau à bouchains ronds est quelque peu plus stable que l'autre dans les deux premières conditions, mais que les deux chavireraient s'ils restaient assez longtemps sur une crête de houle et avancaient à la même vitesse que cette dernière. Le rapport conclut que les programmes du type utilisé pourraient, sous réserve d'une mise au point, servir utilement à l'étude de la stabilité dynamique des navires.

18. Mots-clés
Stabilité, bateaux de pêche, mécanismes de chavirement, simulation par ordinateur.

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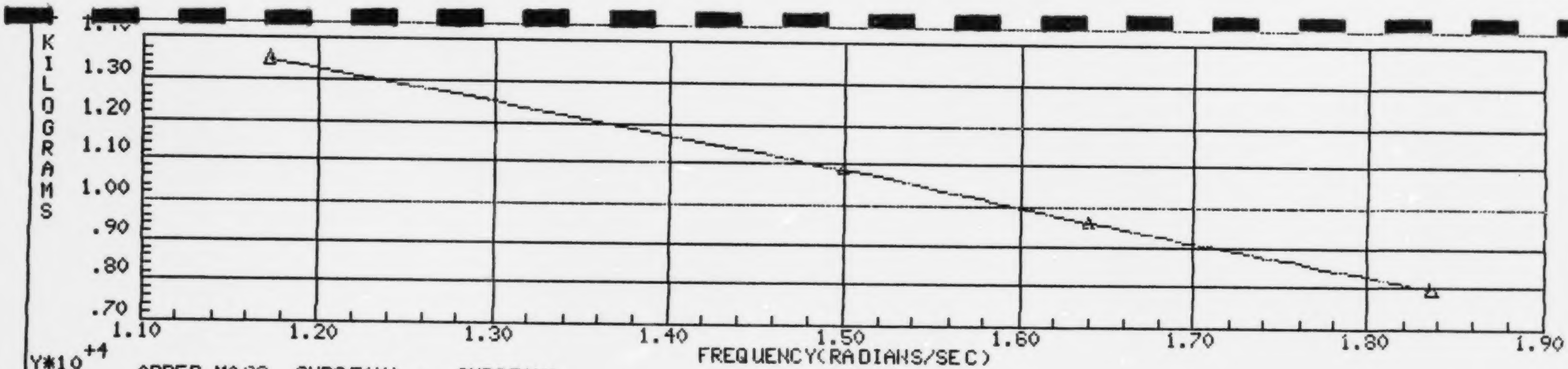
APPENDIX C

AQWA-LINE RESULTS

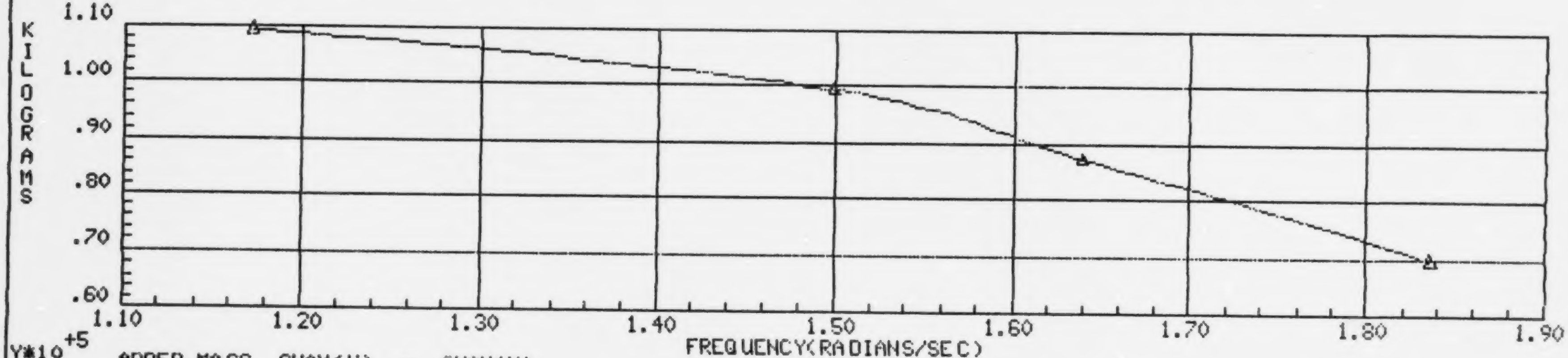
C.1

Added Masses, Radiation Damping and Response Amplitude Operators (RAO's) for " Hard Chine " Vessel

Wave direction 1 = 0 ° (stern sea)
Wave direction 2 = 45 ° (following quartering)
Wave direction 3 = 90 ° (beam sea)
Wave direction 4 = 135 ° (bow quartering)
Wave direction 2 = 180 ° (head sea)



ADDED MASS SURGE(X) - SURGE(X)



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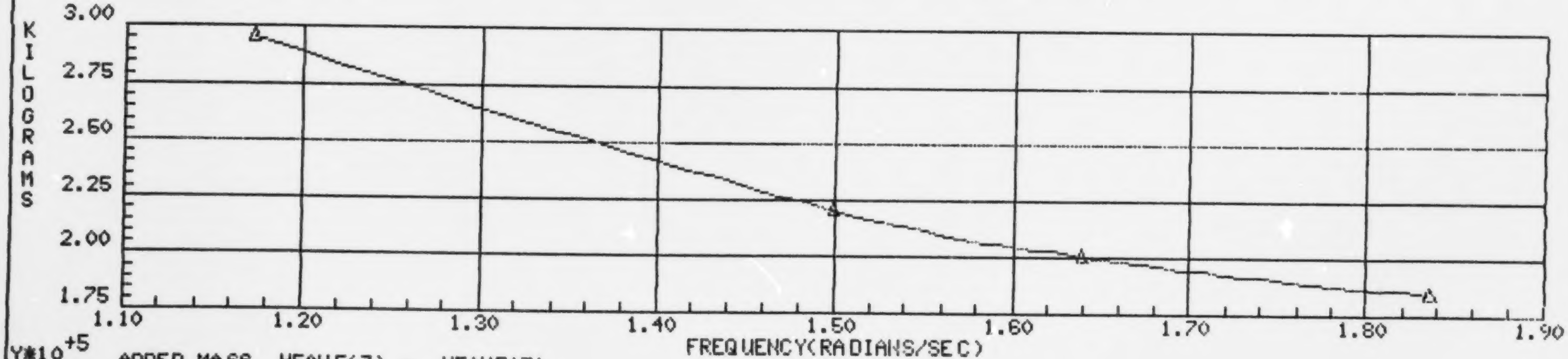

ADDED MASS HEAVE(Z) - HEAVE(Z)

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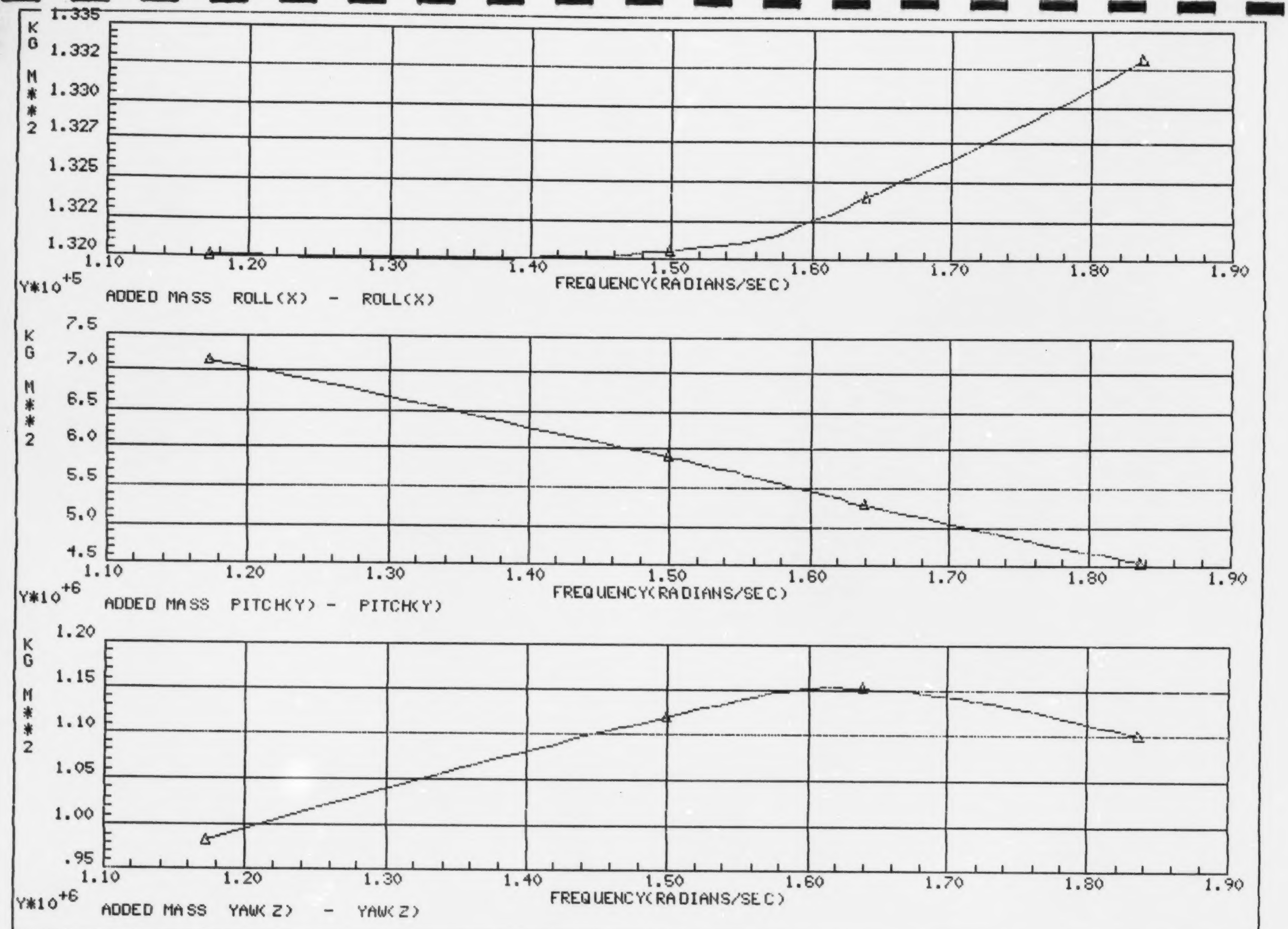
Program-AQWA-LINE - WEST COAST VESSEL



ADDED MASS ROLL(X) - ROLL(X)

ADDED MASS PITCH(Y) - PITCH(Y)

ADDED MASS YAW(Z) - YAW(Z)

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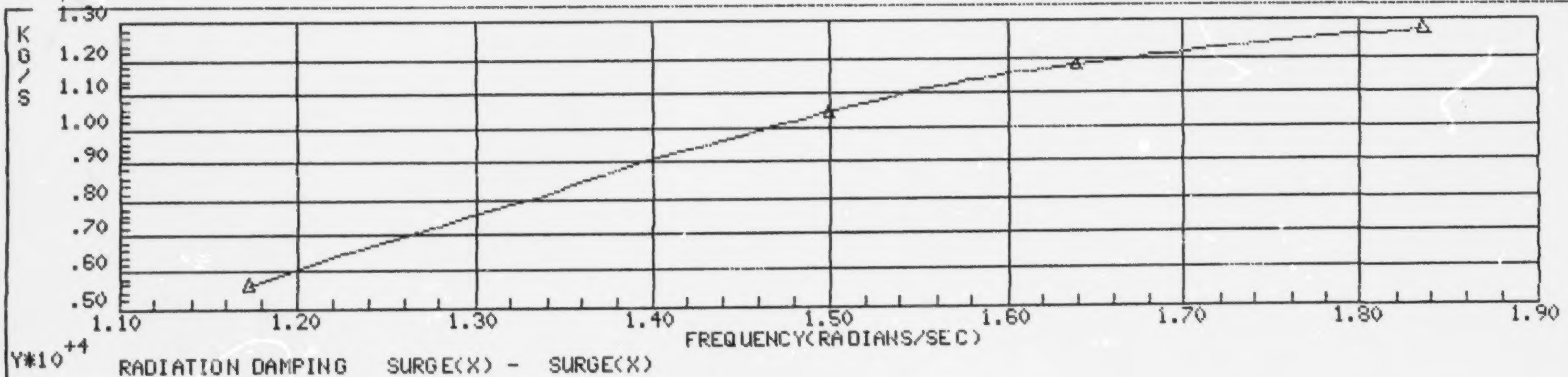

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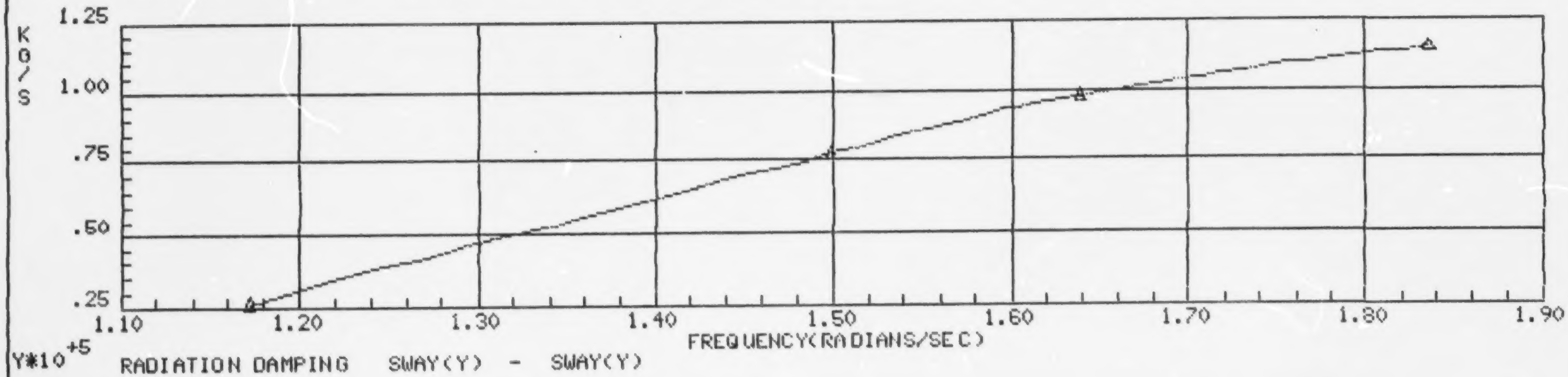

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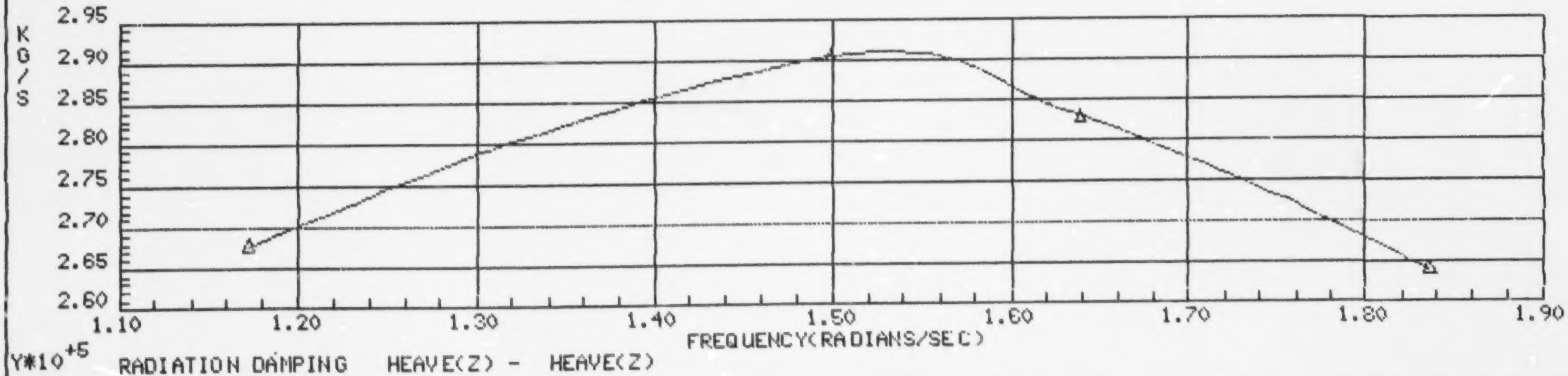

RADIATION DAMPING HEAVE(Z) - HEAVE(Z)

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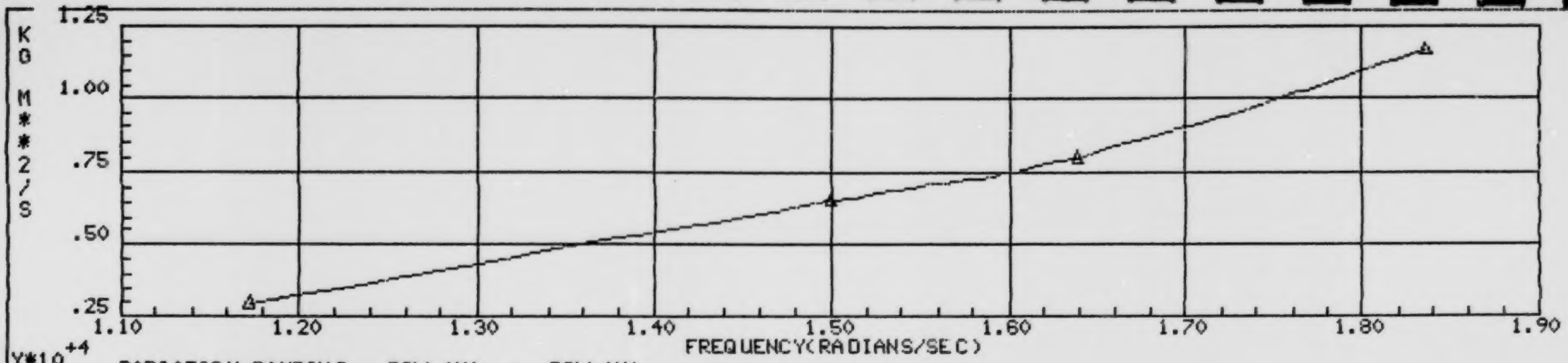

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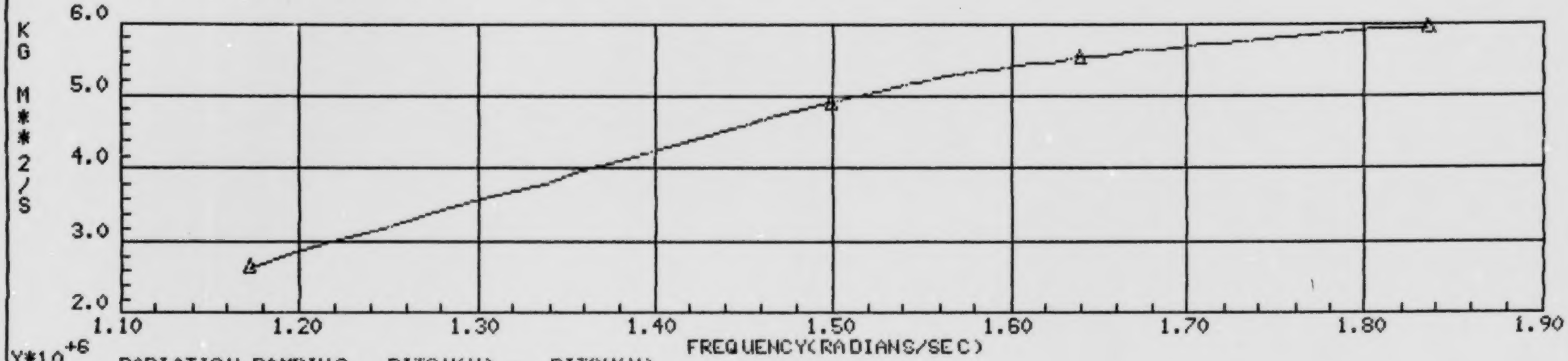

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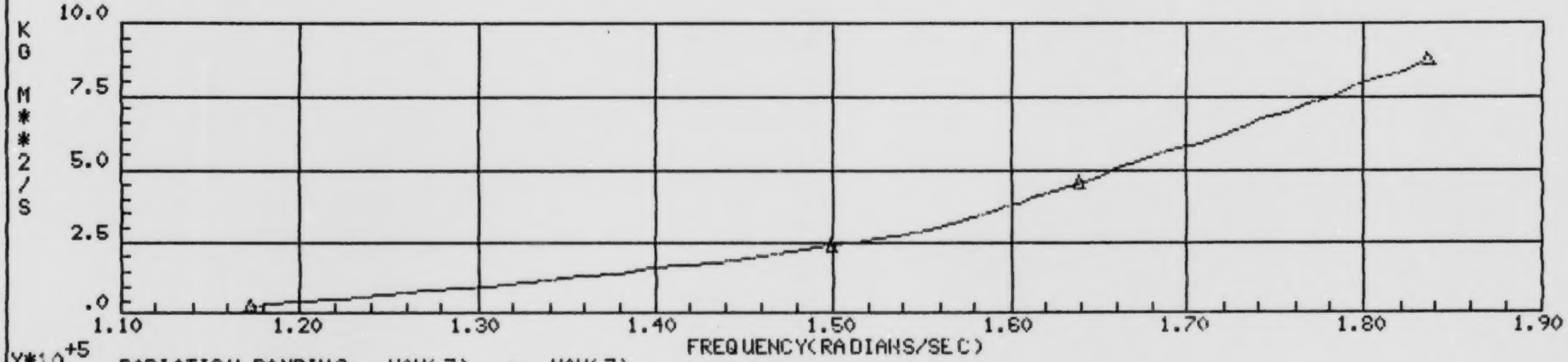

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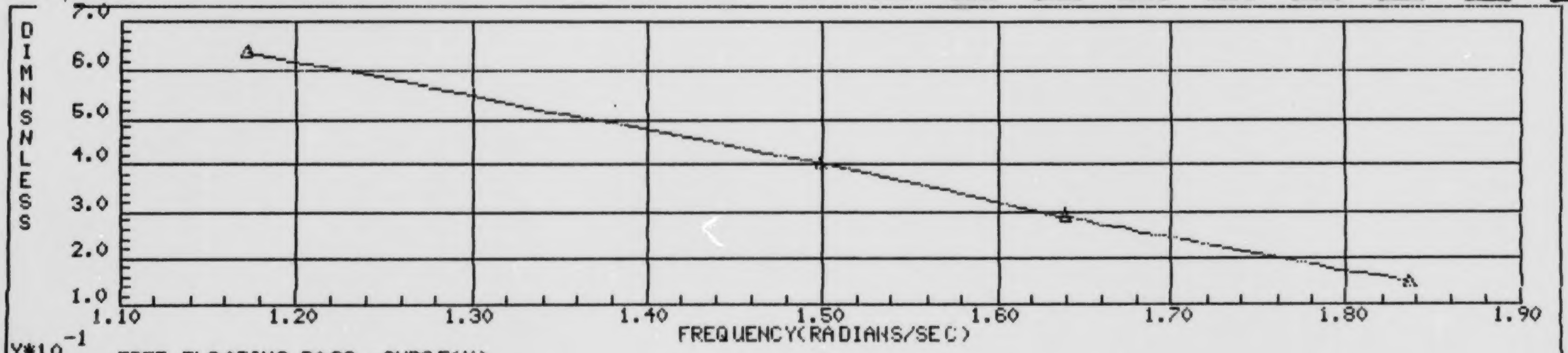

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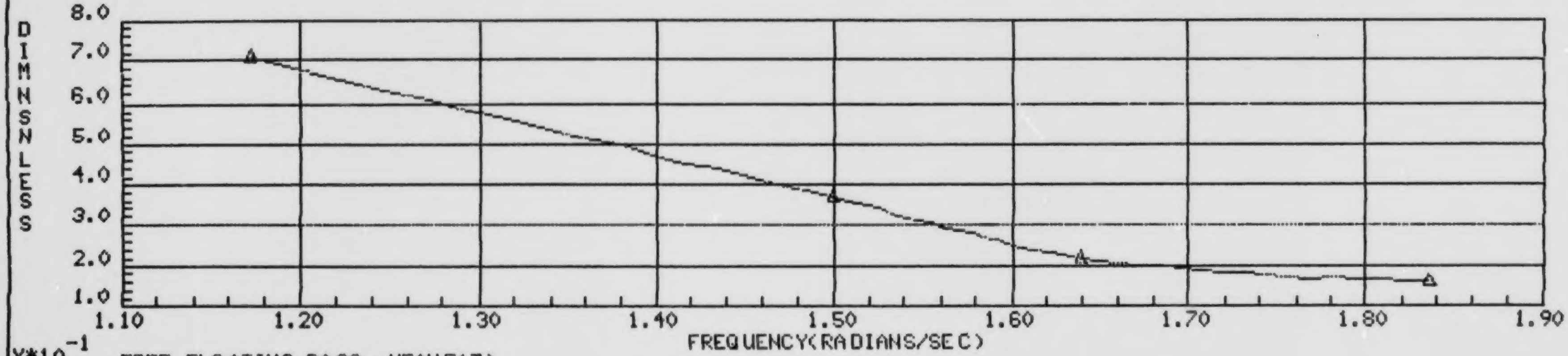

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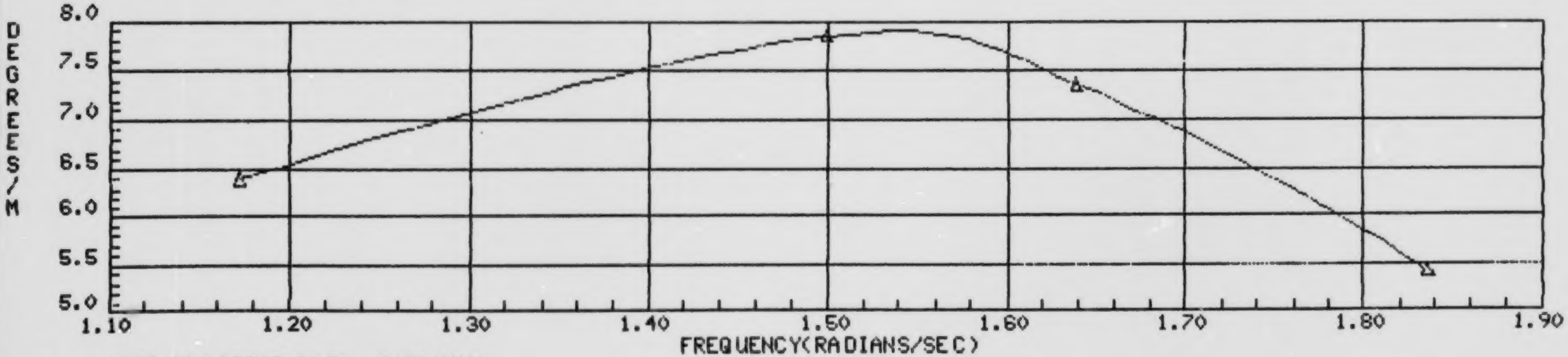

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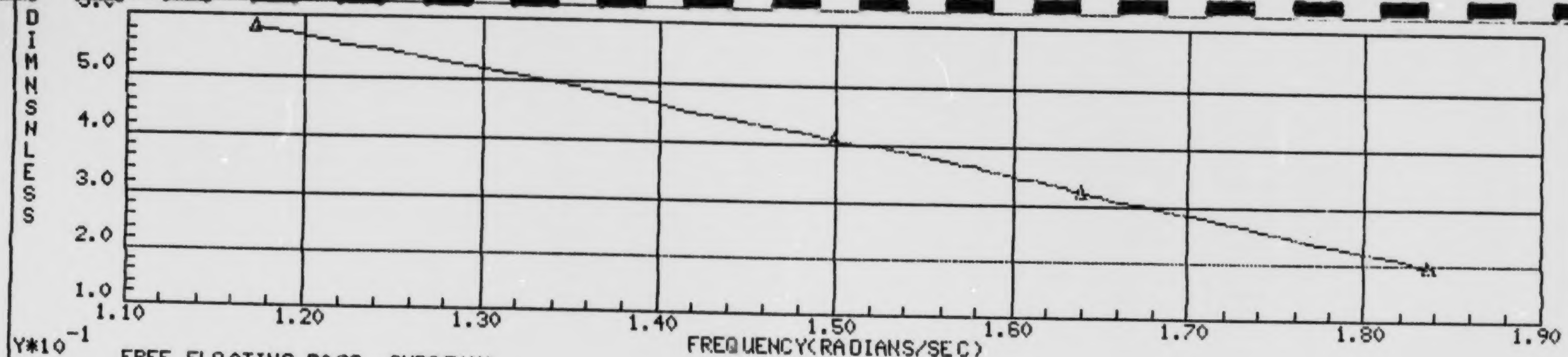

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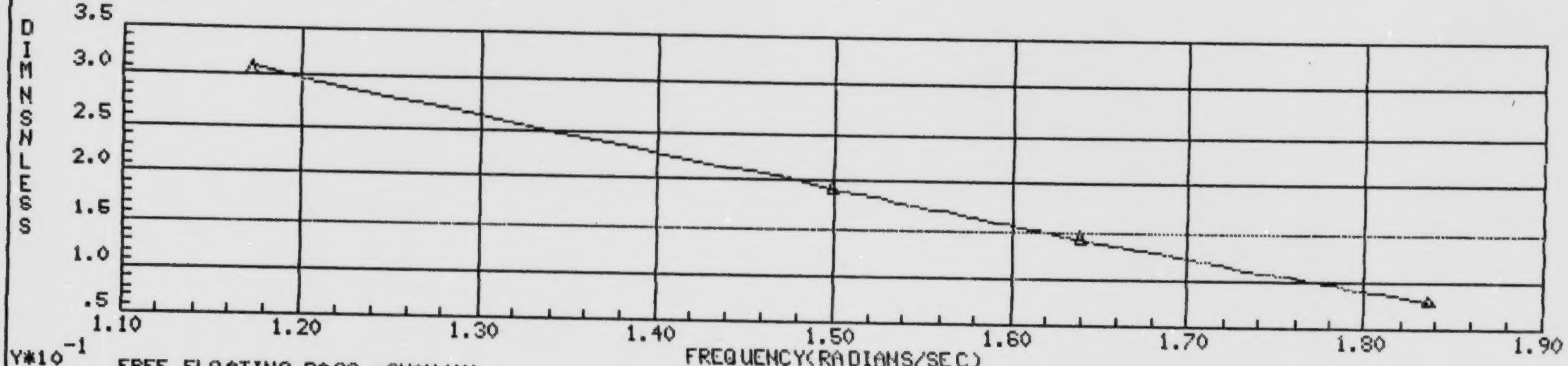

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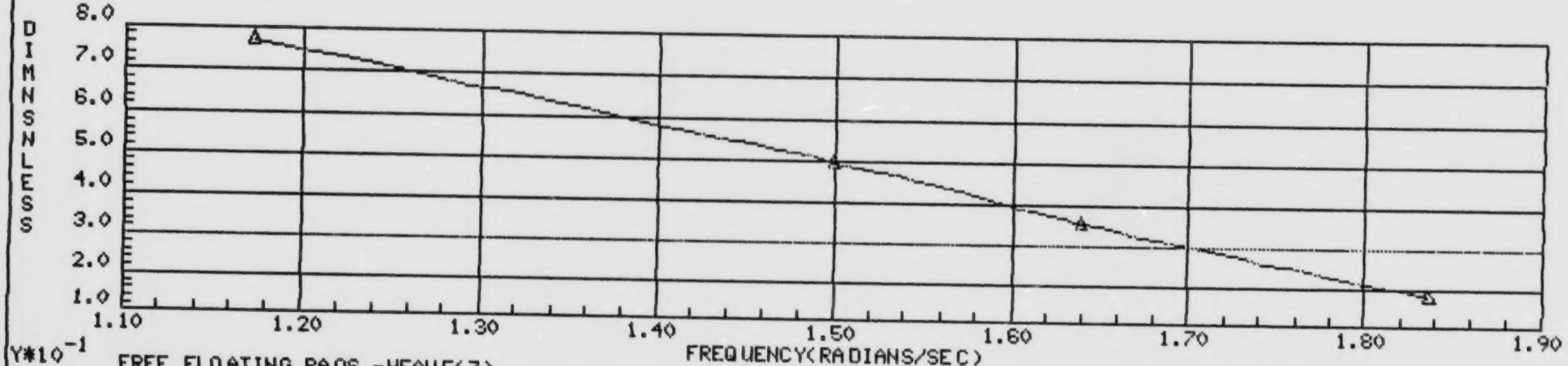

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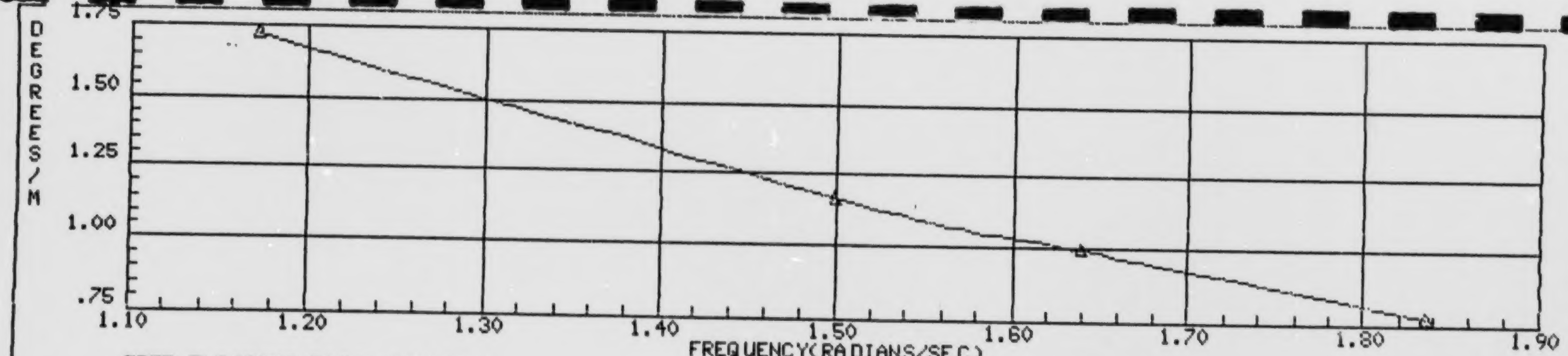

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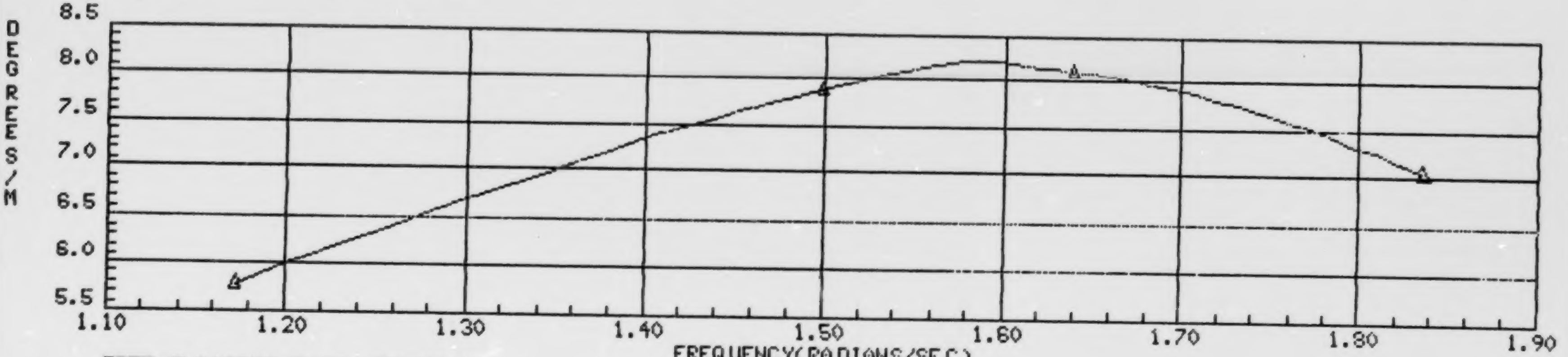

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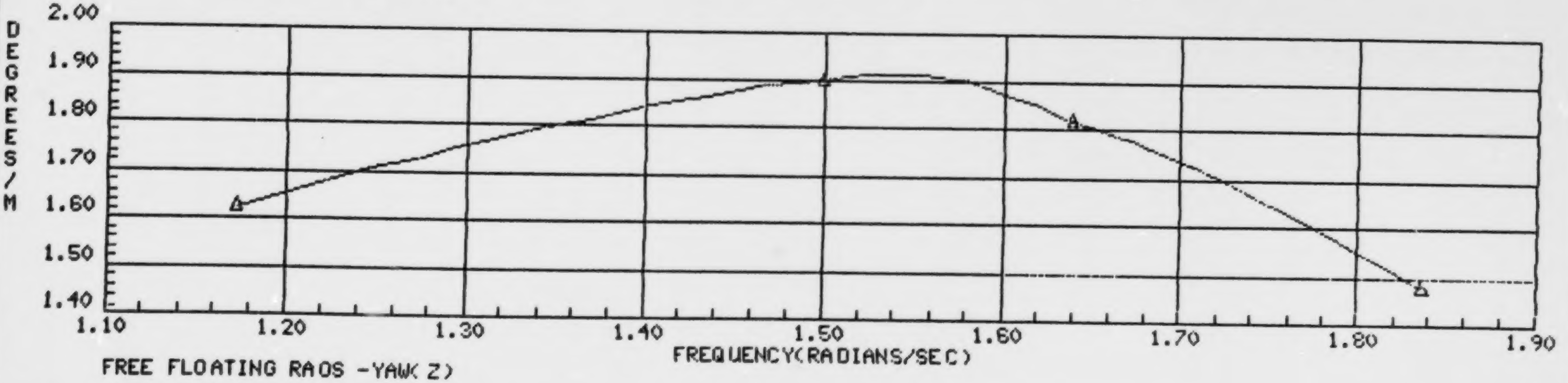

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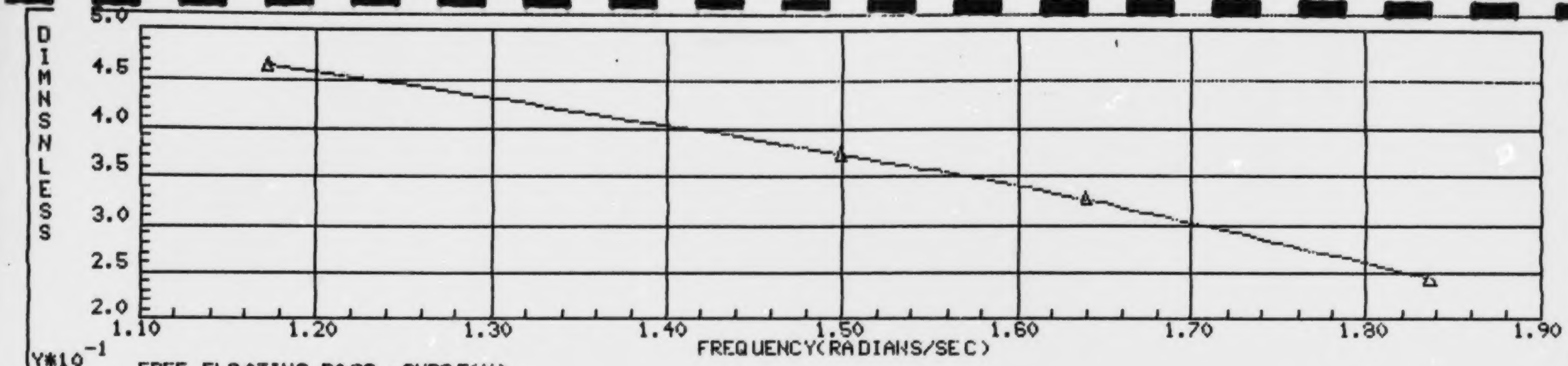

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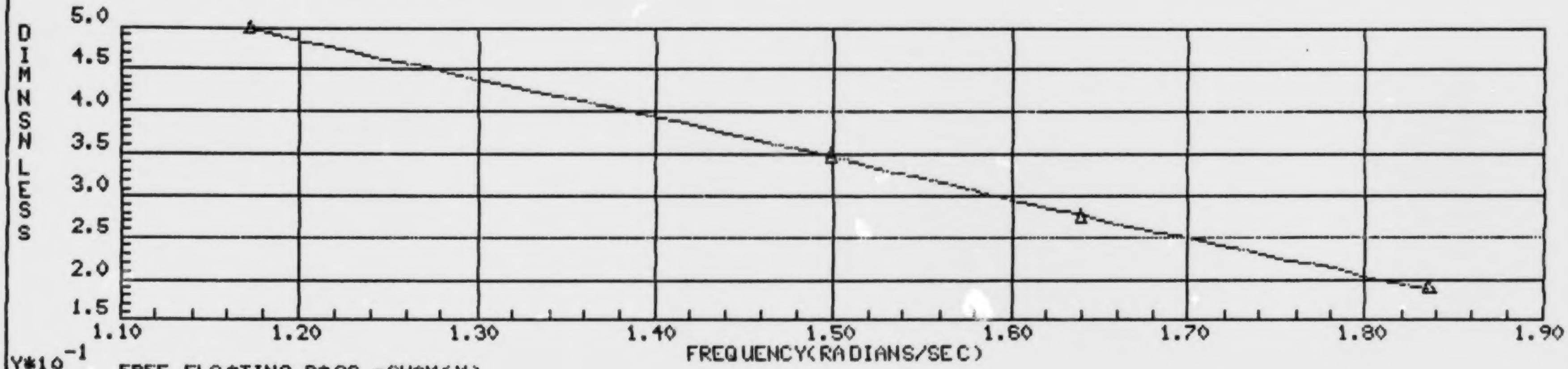

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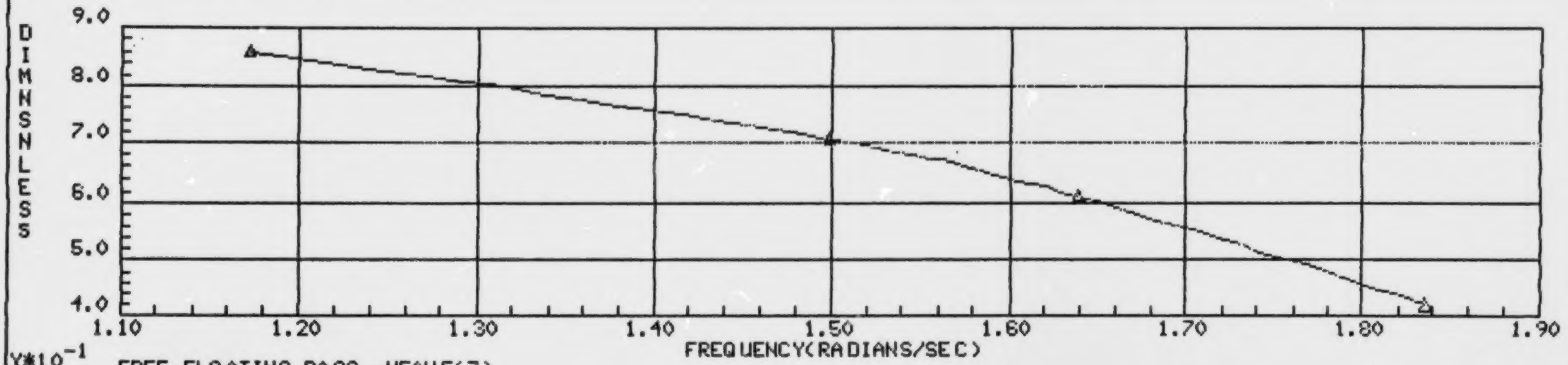

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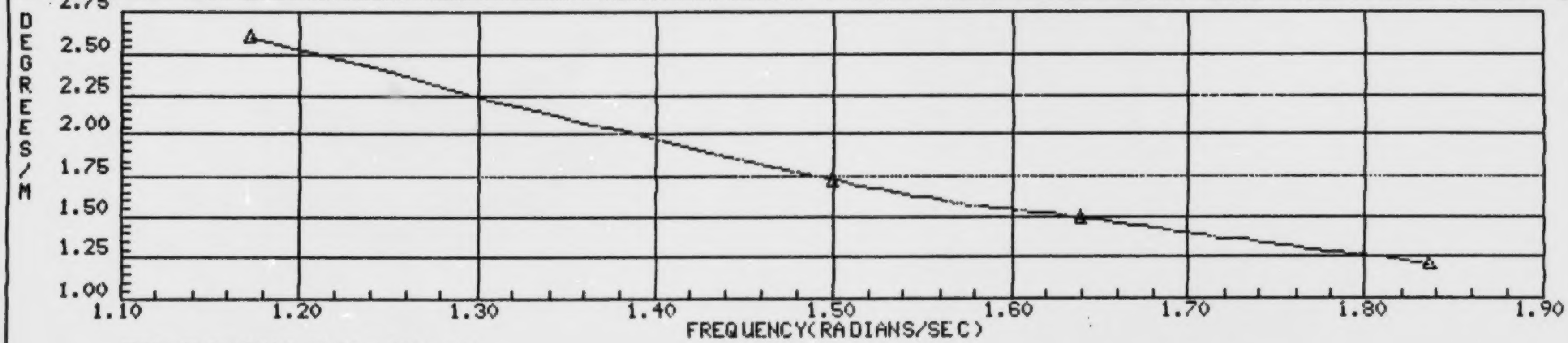

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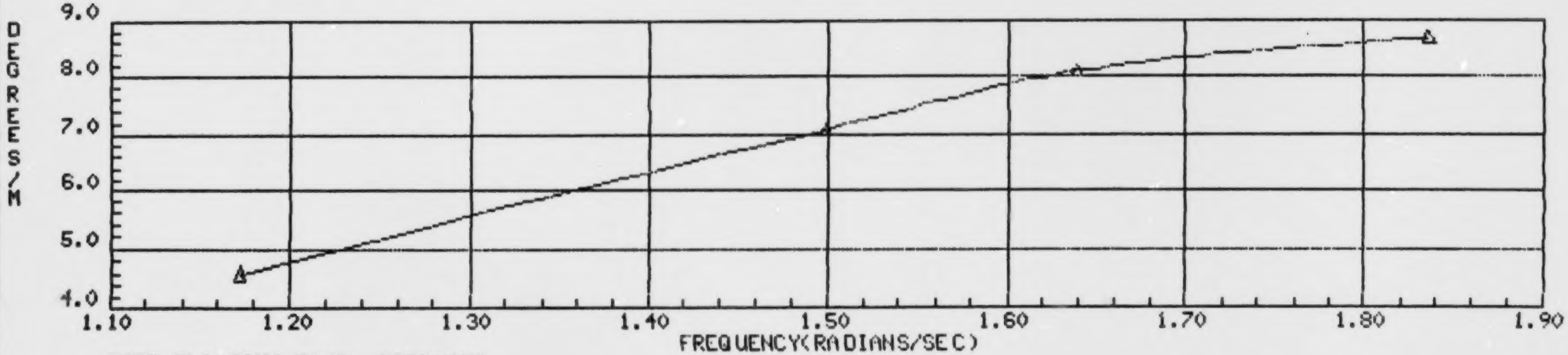

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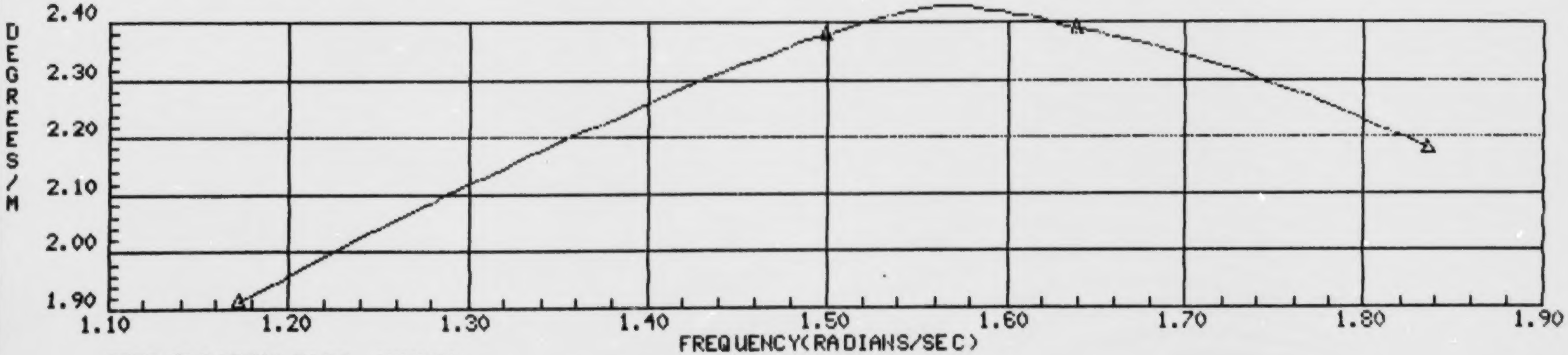

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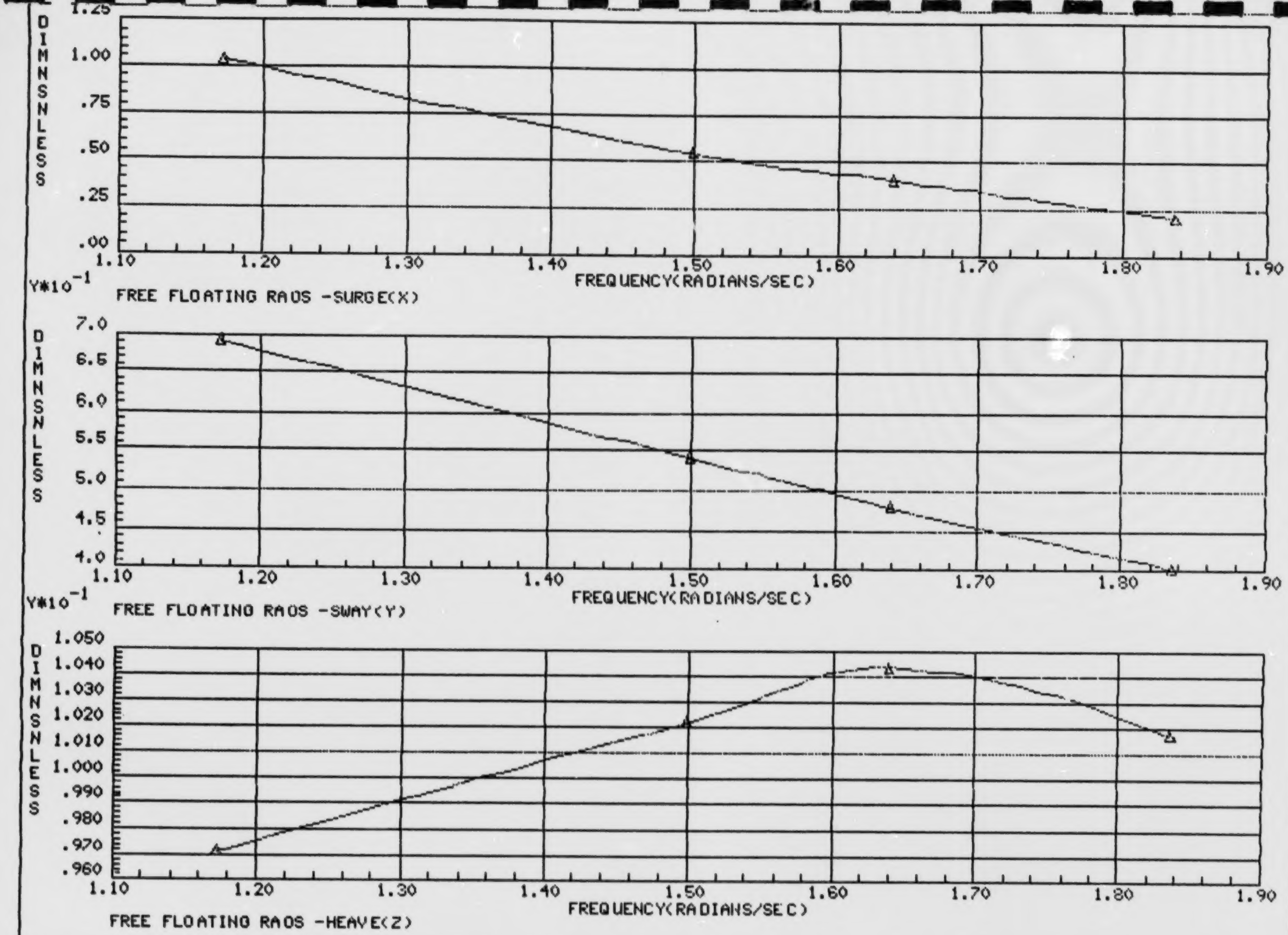

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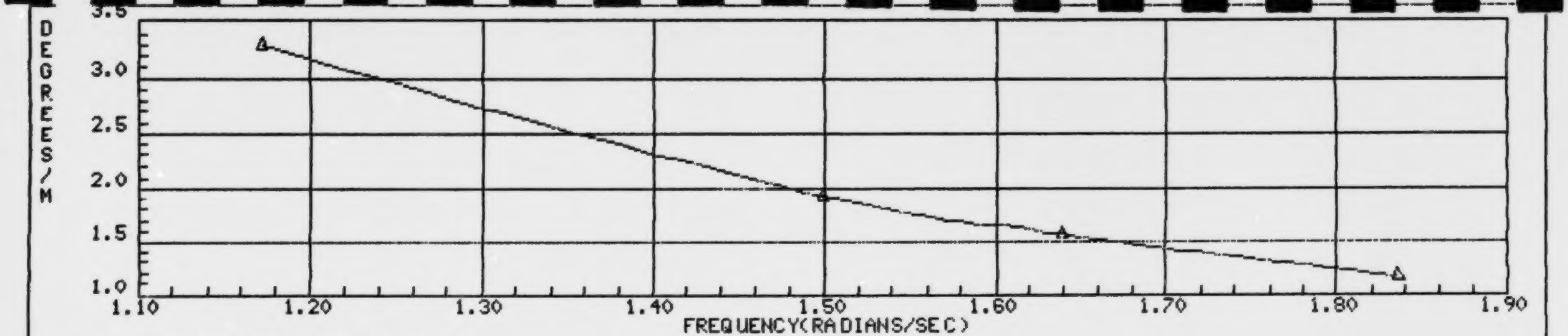

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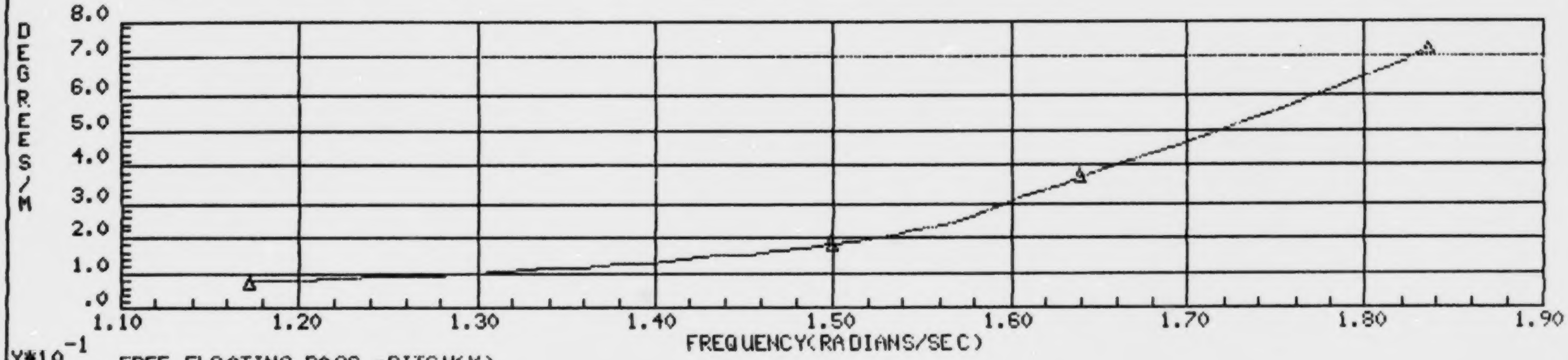

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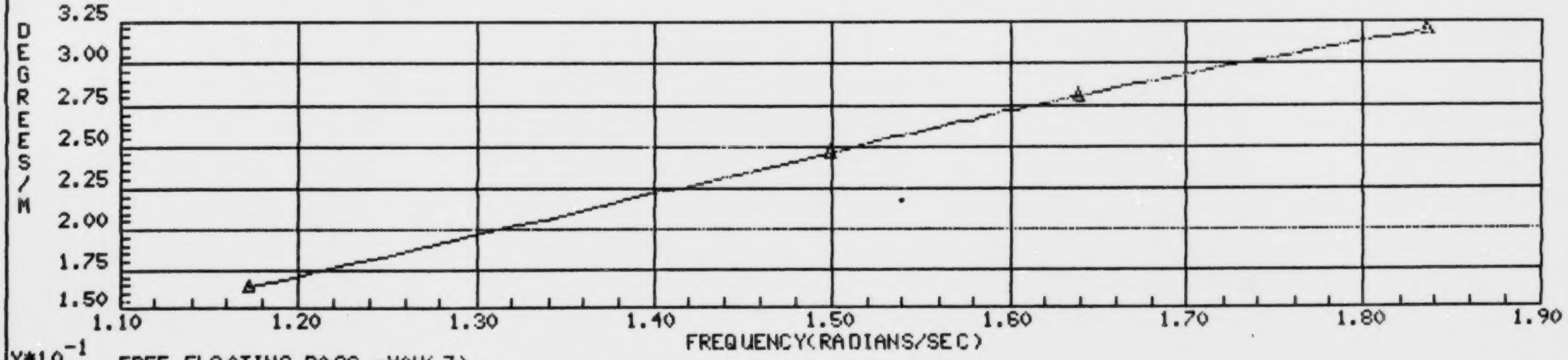

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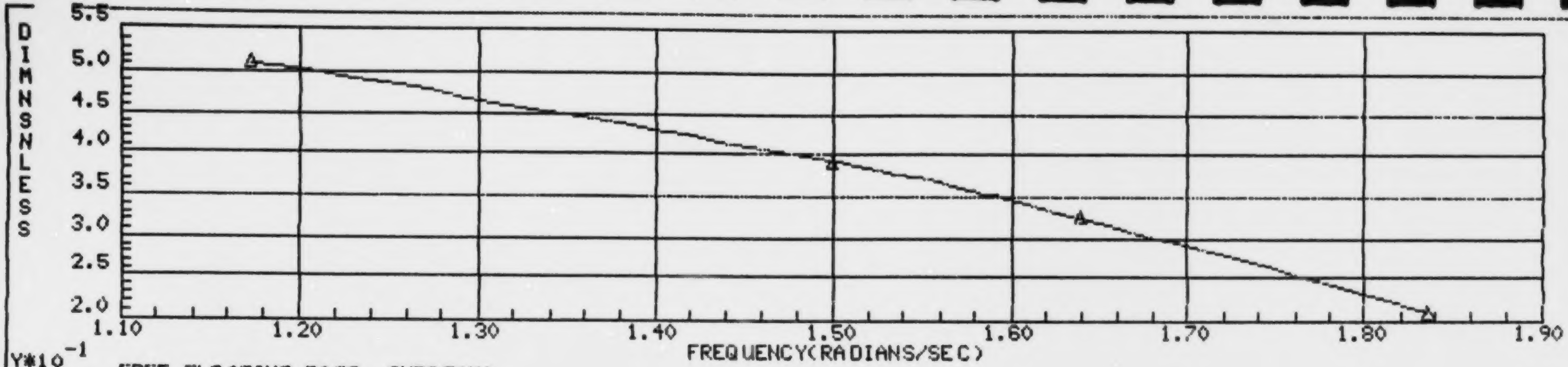

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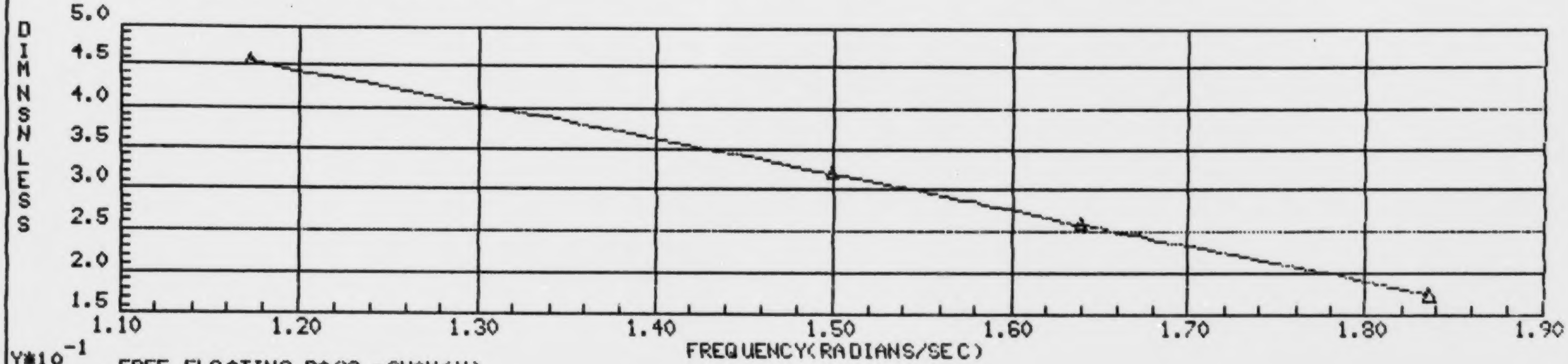

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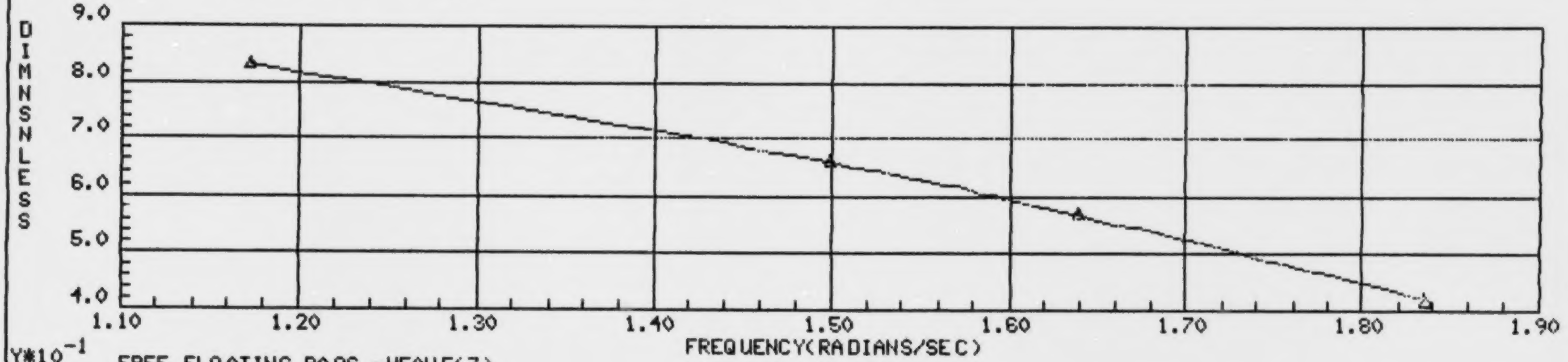

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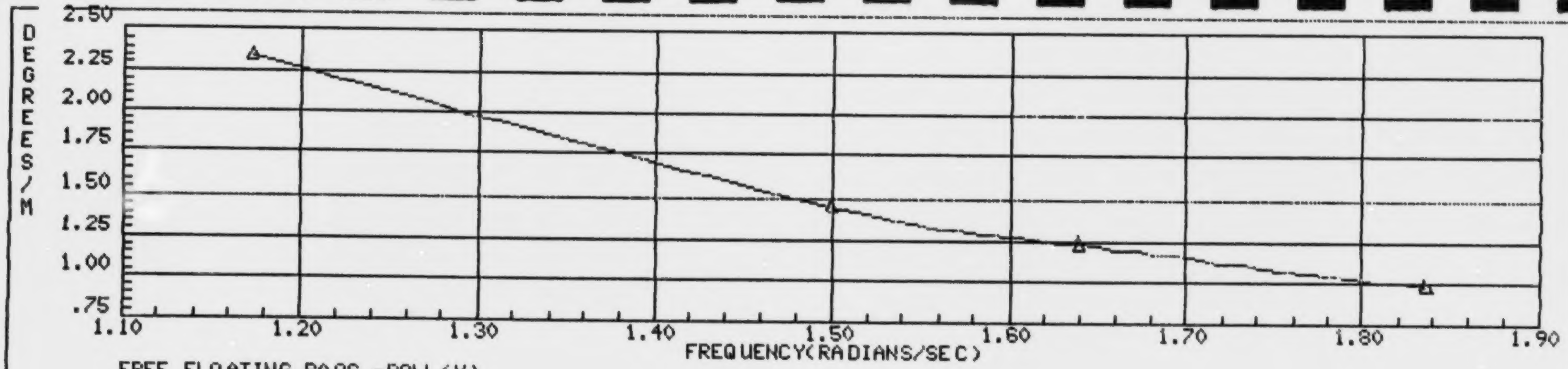

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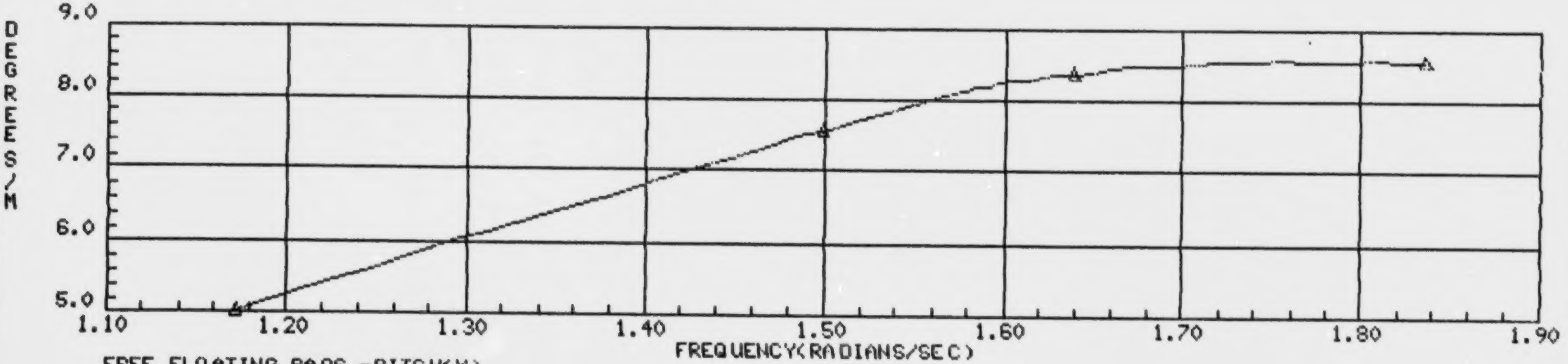
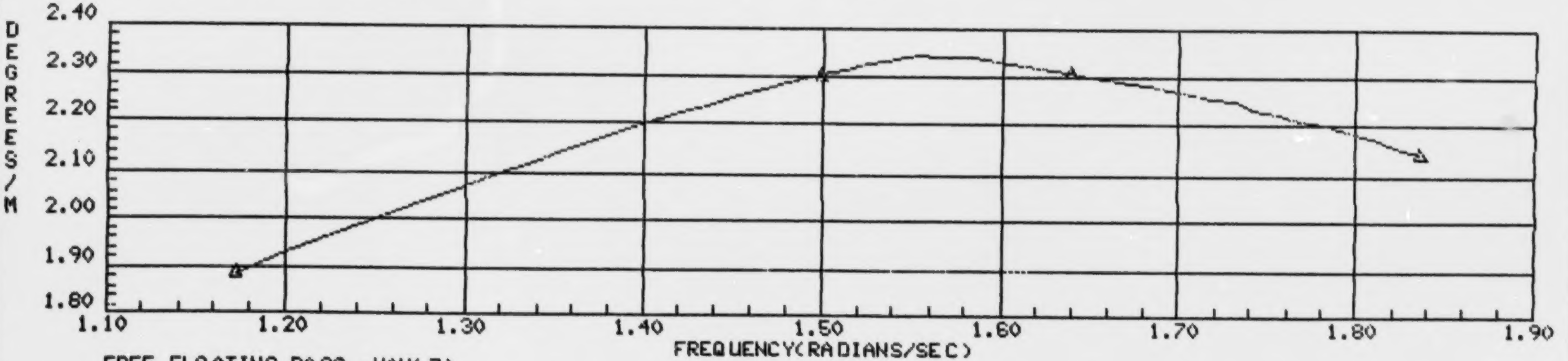


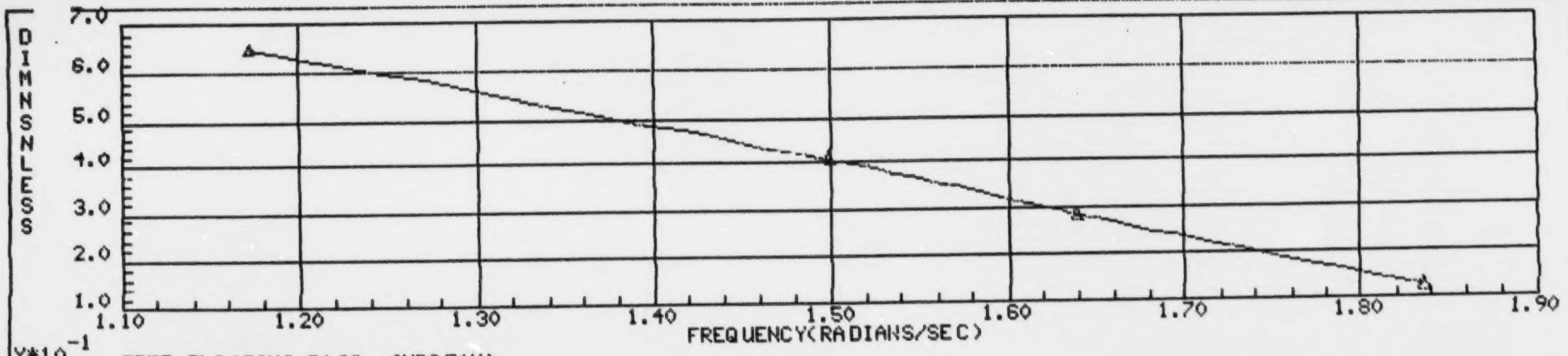
FREE FLOATING RAOS -PITCH(Y)



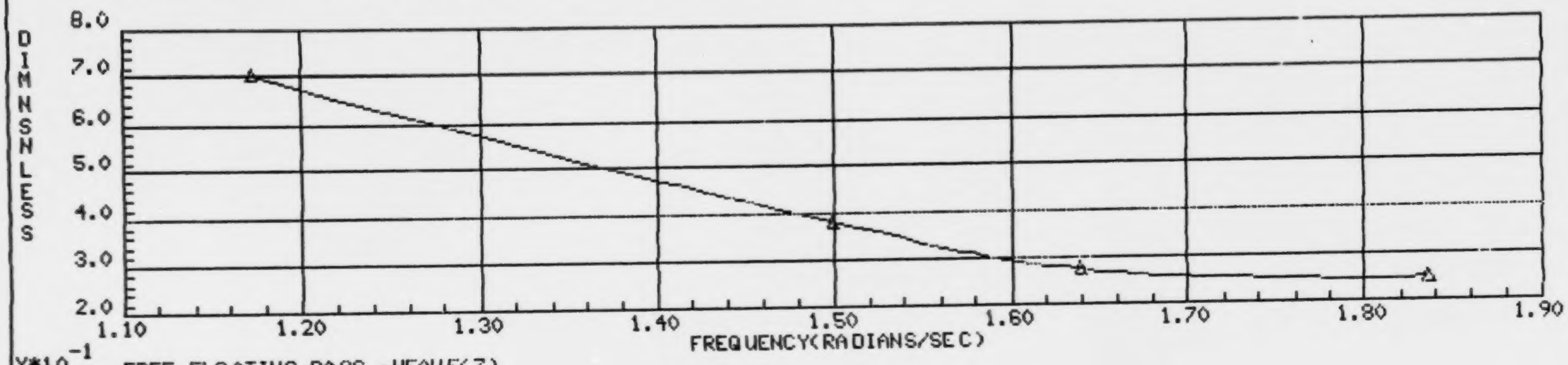
FREE FLOATING RAOS -YAW(Z)

STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 5

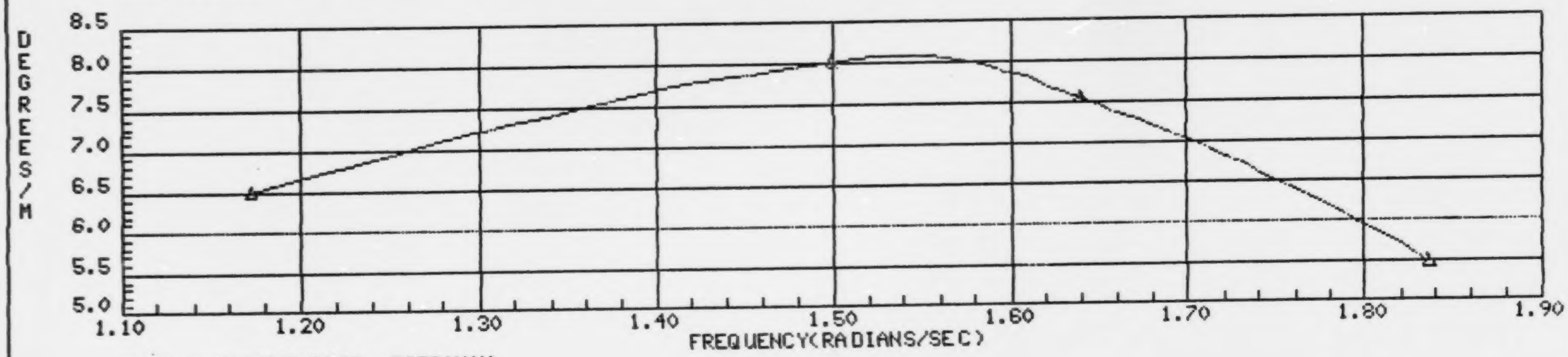
Progran-AQWA-LINE - WEST COAST VESSEL



FREE FLOATING RAOS -SURGE(X)



FREE FLOATING RAOS -HEAVE(Z)



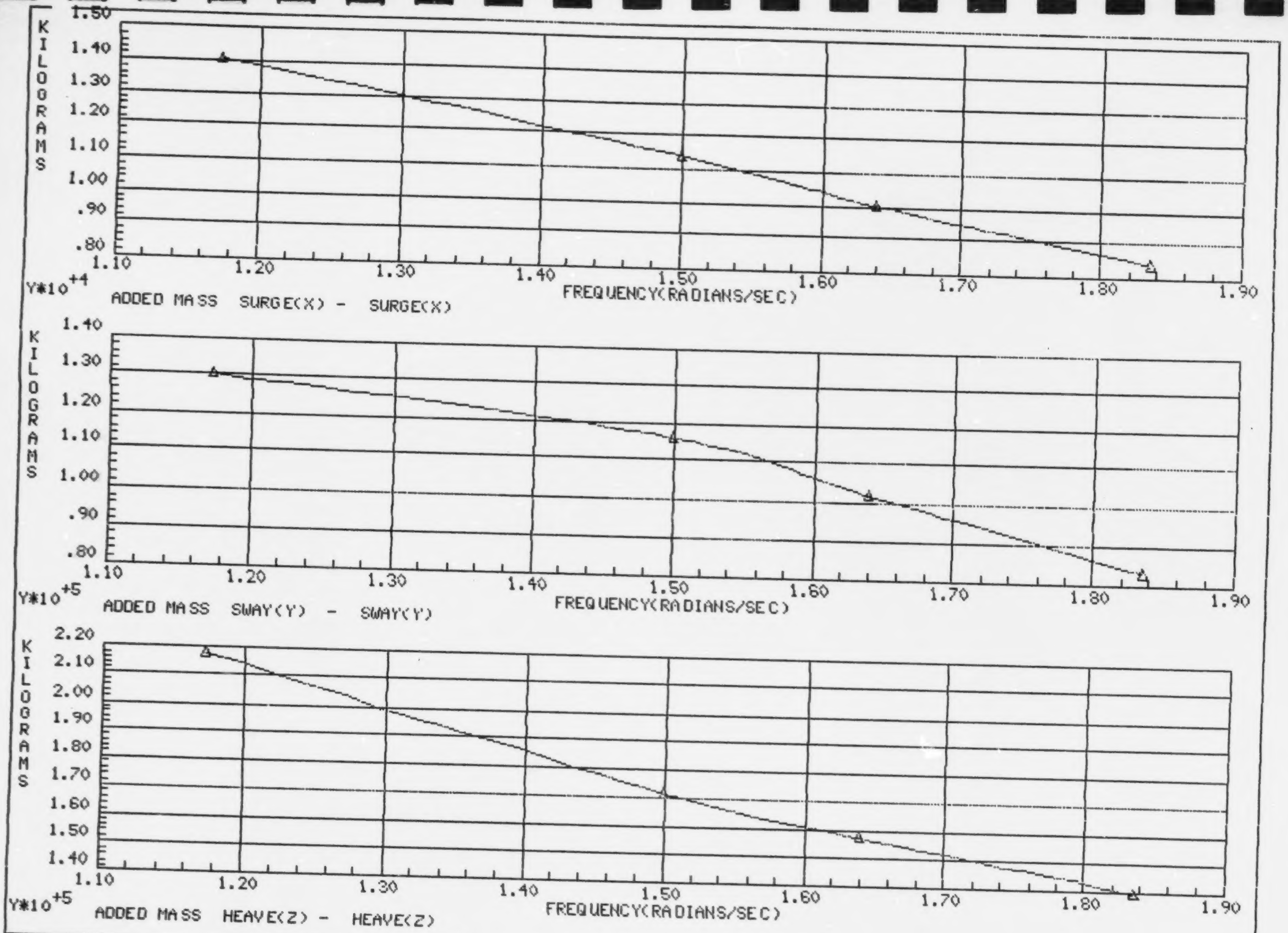
FREE FLOATING RAOS -PITCH(Y)

STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 6
Progran-AQWA-LINE - WEST COAST VESSEL

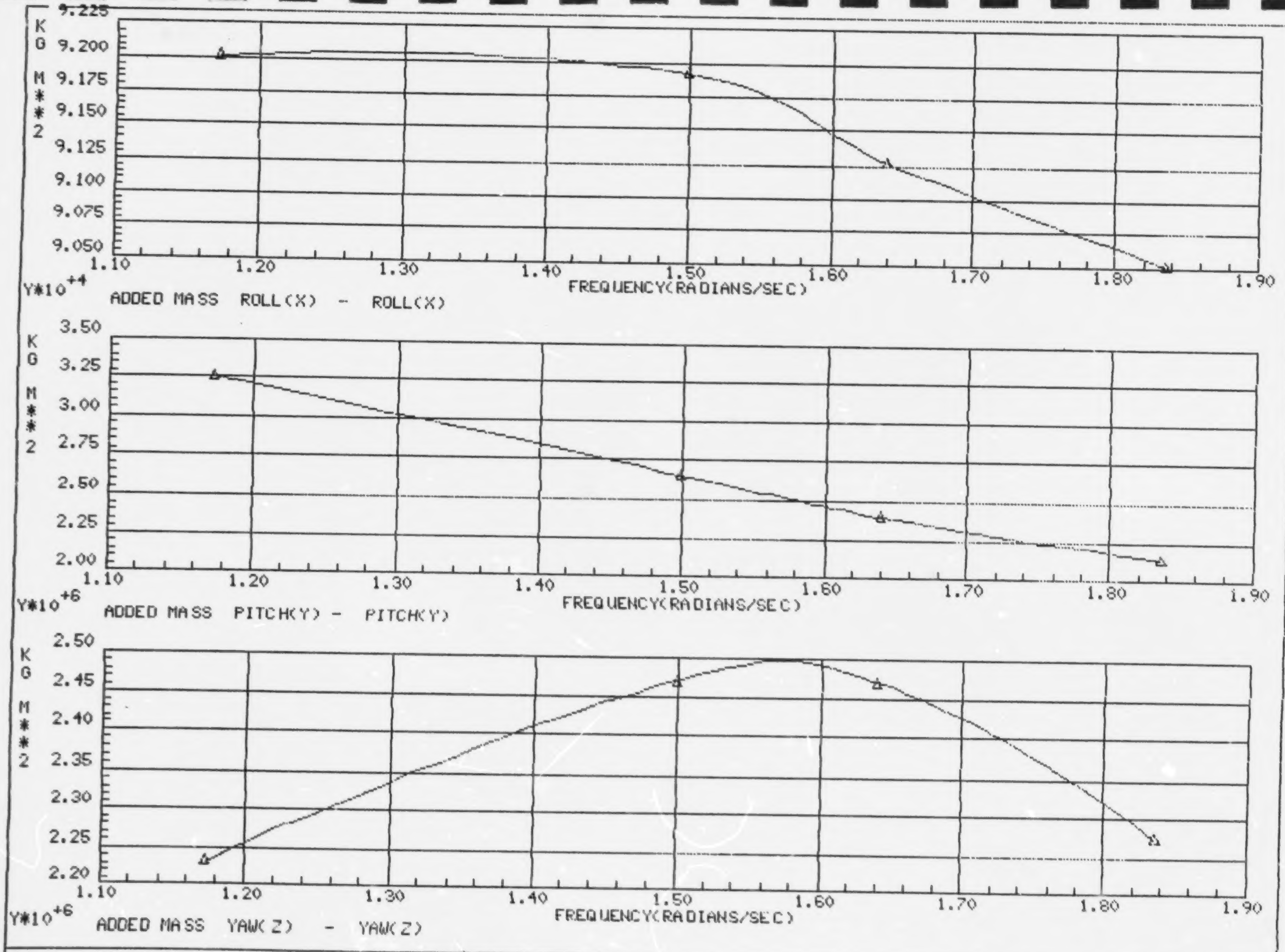
C.2

Added Masses, Radiation Damping and Response Amplitude Operators (RAO's) for " Round Bilge " Vessel

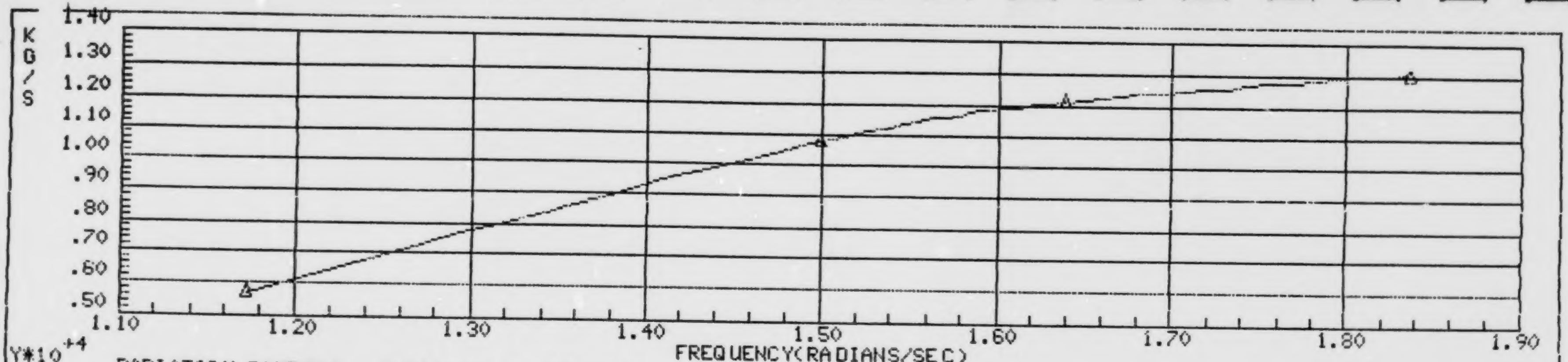
Wave direction 1 = 0 ° (stern sea)
Wave direction 2 = 45 ° (following quartering)
Wave direction 3 = 90 ° (beam sea)
Wave direction 4 = 135 ° (bow quartering)
Wave direction 2 = 180 ° (head sea)



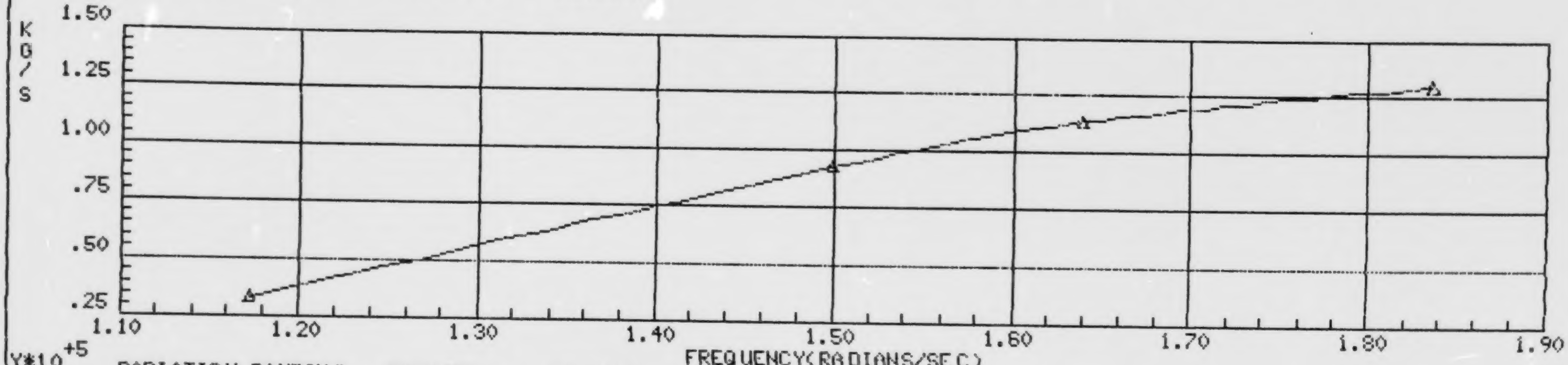
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 1
Program-AQWA-LINE - EAST COAST VESSEL



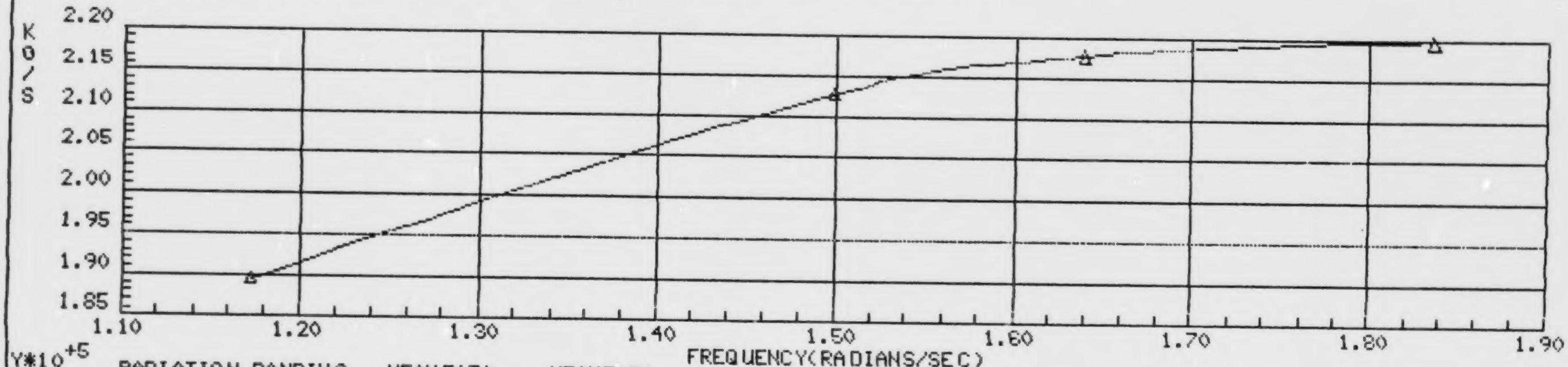
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 1
Progran-AQWA-LINE - EAST COAST VESSEL



RADIATION DAMPING SURGE(X) - SURGE(X)

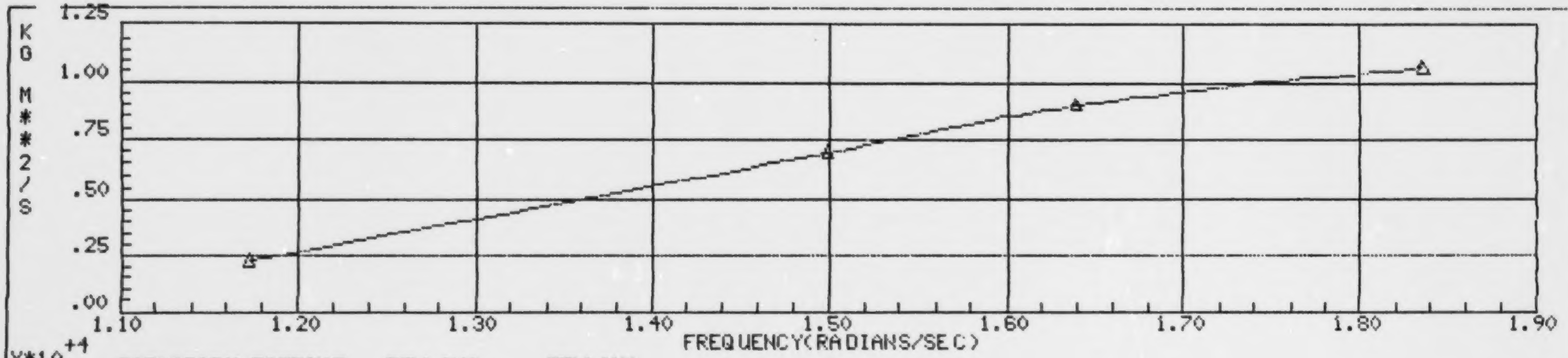


RADIATION DAMPING SWAY(Y) - SWAY(Y)

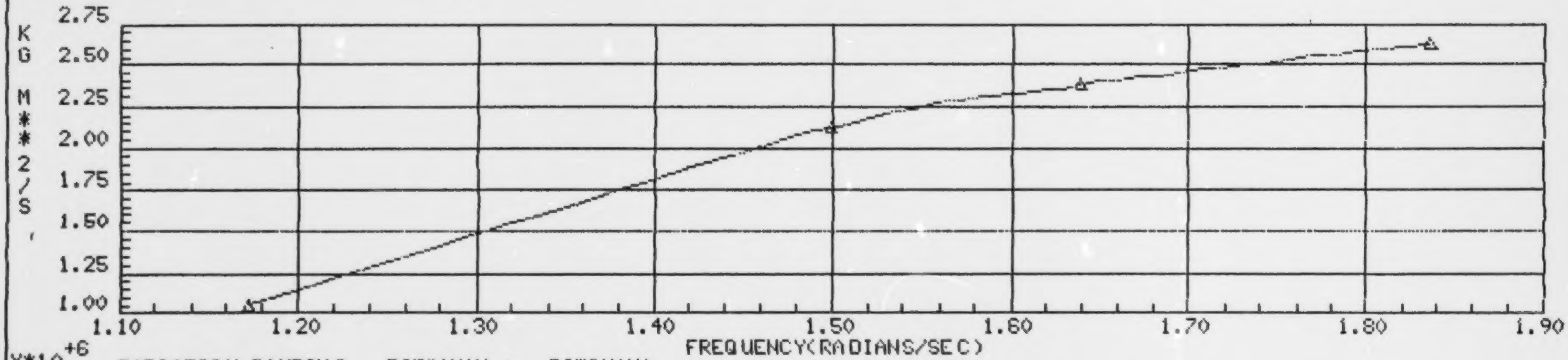


RADIATION DAMPING HEAVE(Z) - HEAVE(Z)

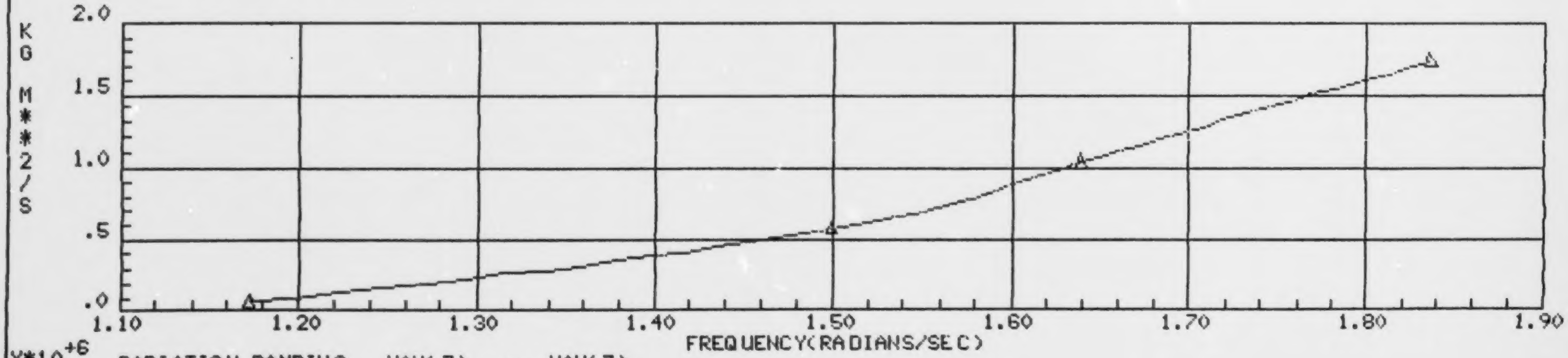
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 1
Program-AQWA-LINE - EAST COAST VESSEL



RADIATION DAMPING ROLL(X) - ROLL(X)

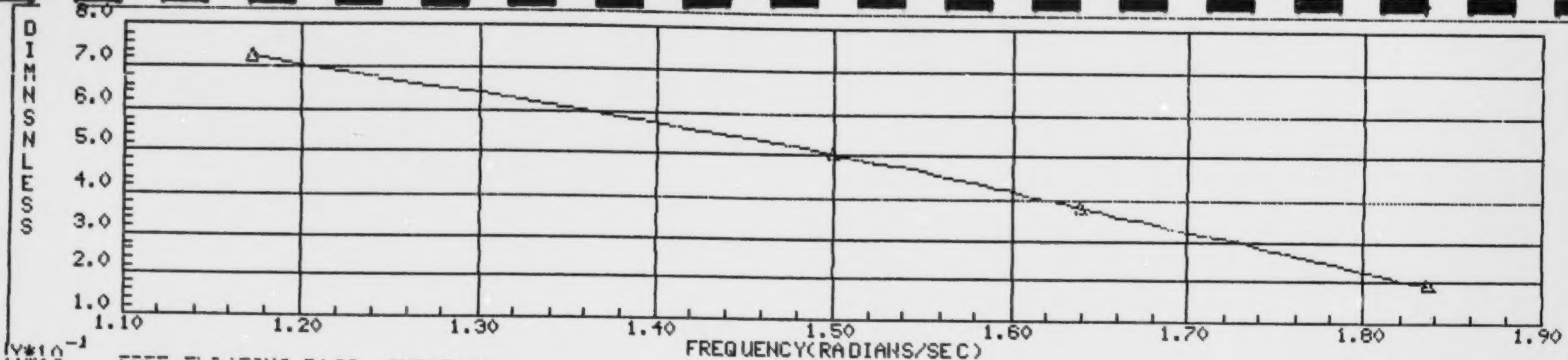


RADIATION DAMPING PITCH(Y) - PITCH(Y)

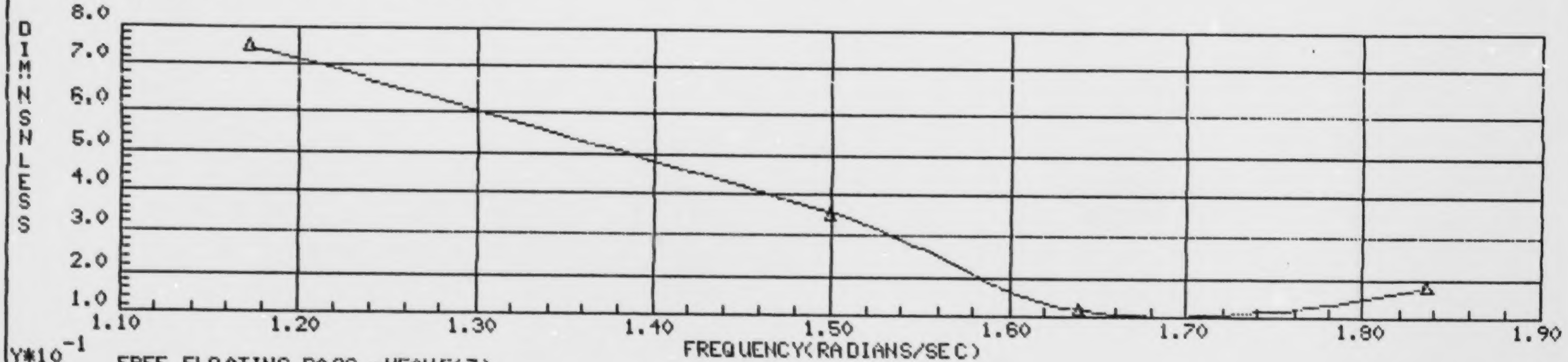


RADIATION DAMPING YAW(Z) - YAW(Z)

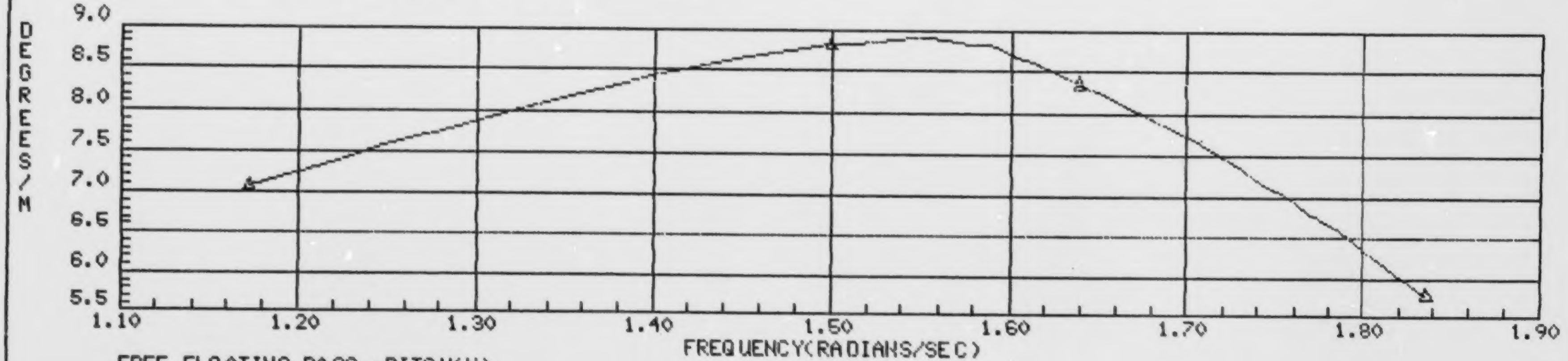
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 1
Progran-AQWA-LINE - EAST COAST VESSEL



FREE FLOATING RAOS -SURGE(X)

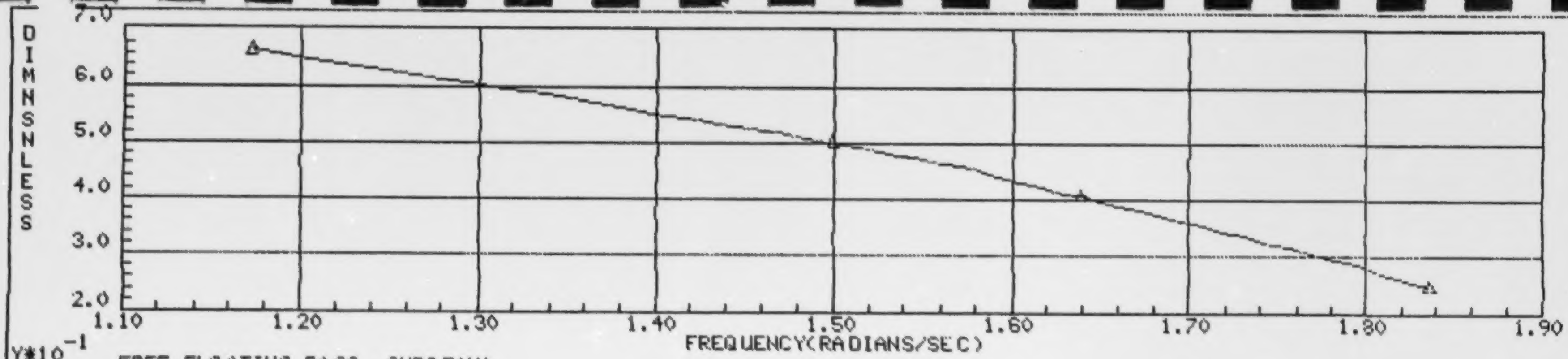


FREE FLOATING RAOS -HEAVE(Z)

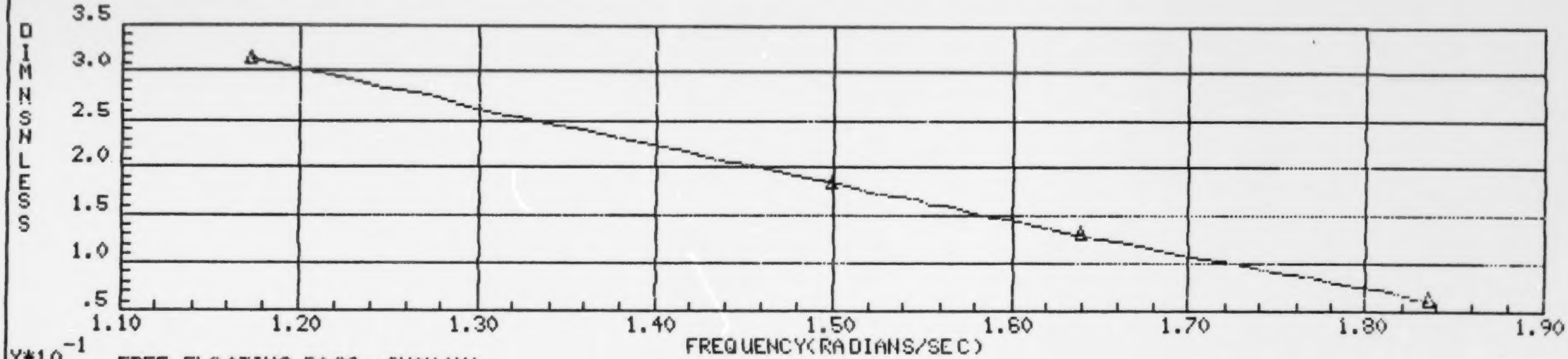


FREE FLOATING RAOS -PITCH(Y)

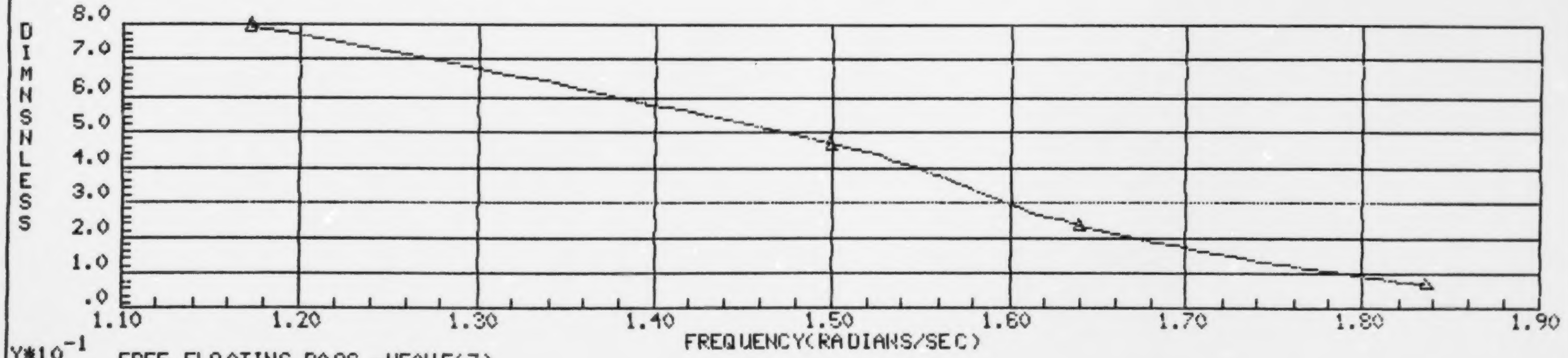
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 1
Program-AQWA-LINE - EAST COAST VESSEL



FREE FLOATING RAOS -SURGE(X)

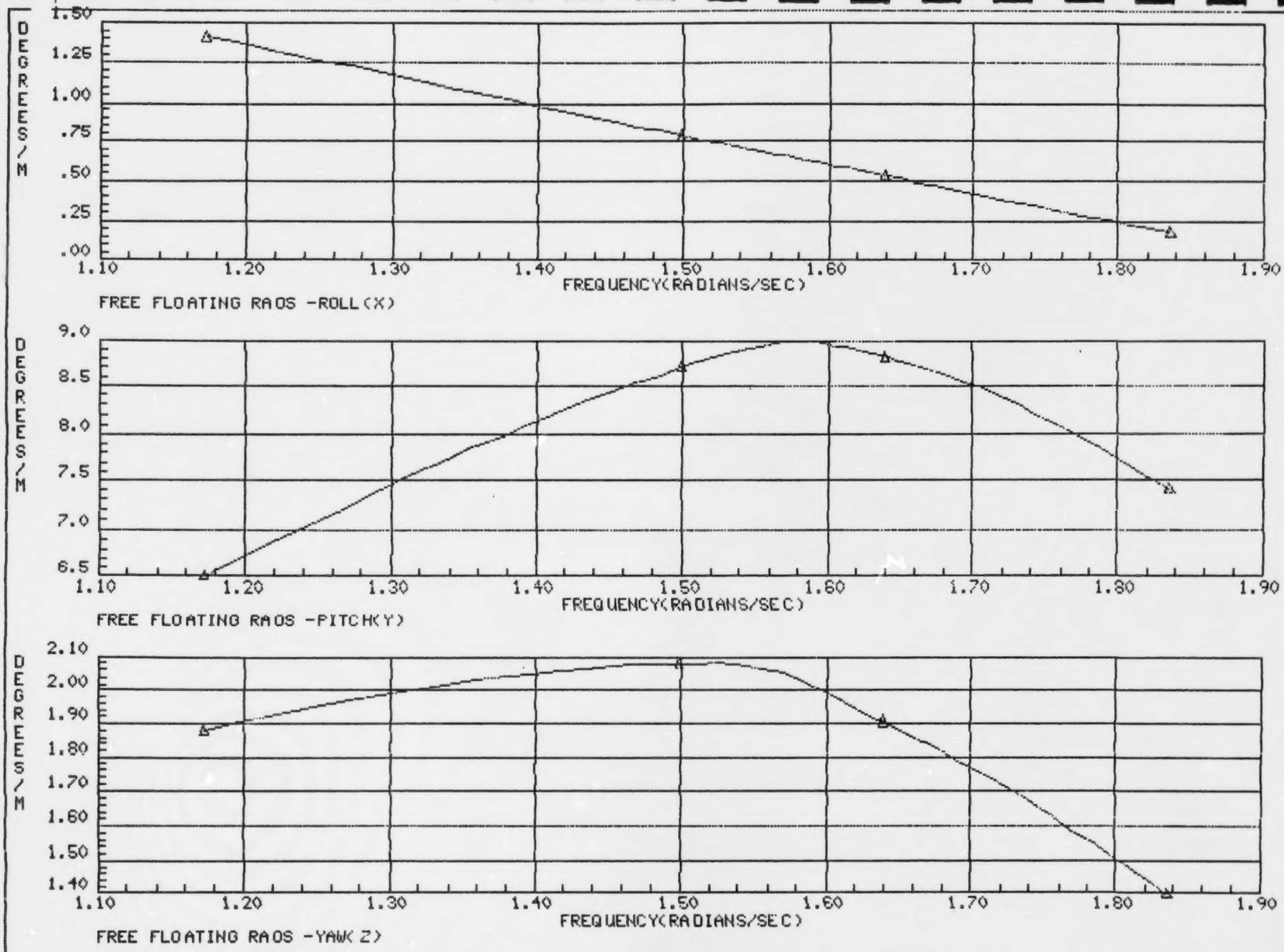


FREE FLOATING RAOS -SWAY(Y)



FREE FLOATING RAOS -HEAVE(Z)

STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 2
Program-AQWA-LINE - EAST COAST VESSEL

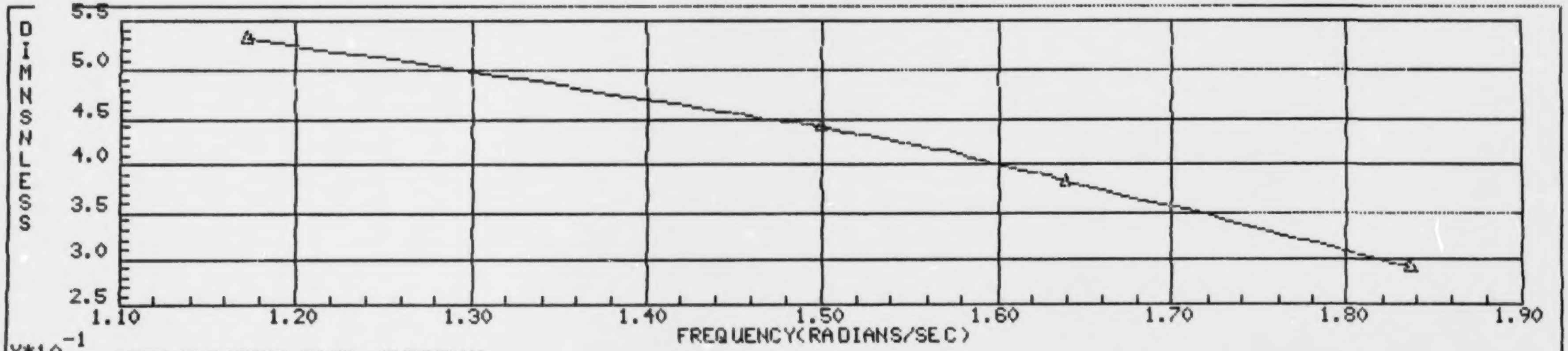


FREE FLOATING RAOS -ROLL(X)

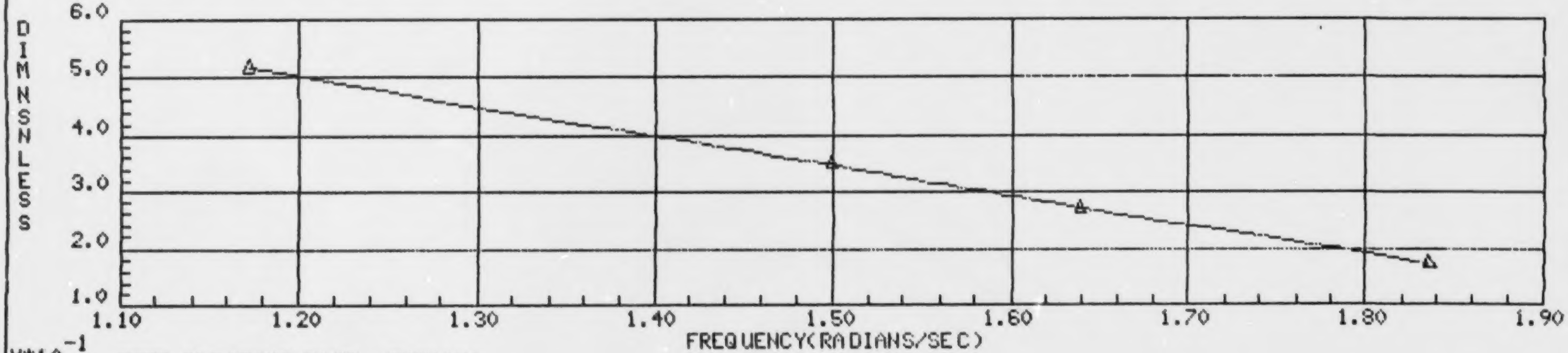
FREE FLOATING RAOS -PITCH(Y)

FREE FLOATING RAOS -YAWK Z)

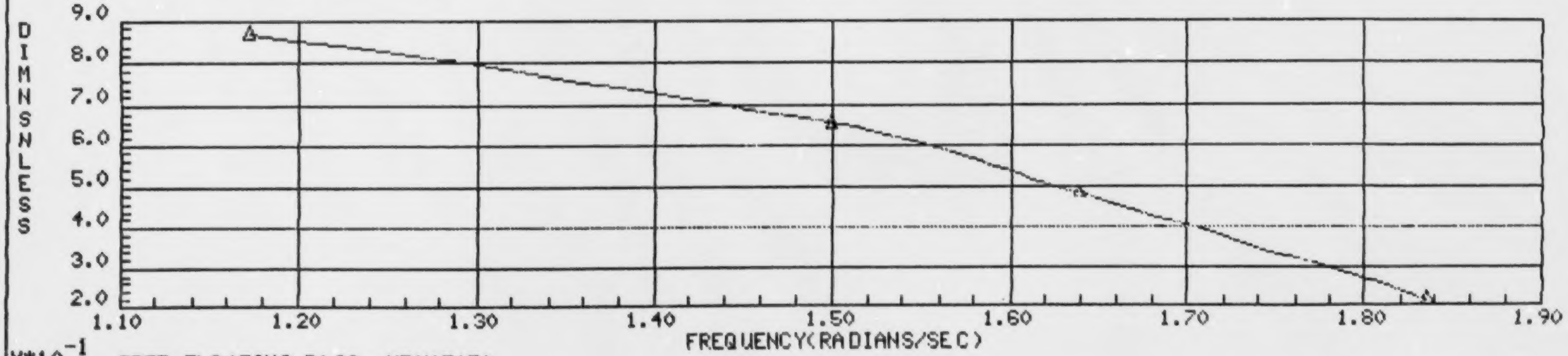
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 2
Program-AQWA-LINE - EAST COAST VESSEL



Y*10^{-1} FREE FLOATING RAOS -SURGE(X)

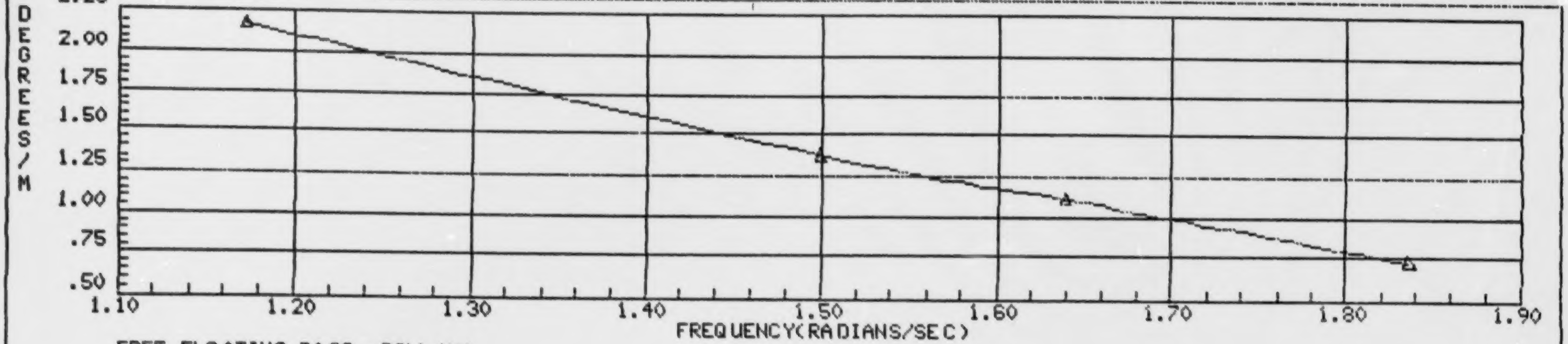


Y*10^{-1} FREE FLOATING RAOS -SWAY(Y)

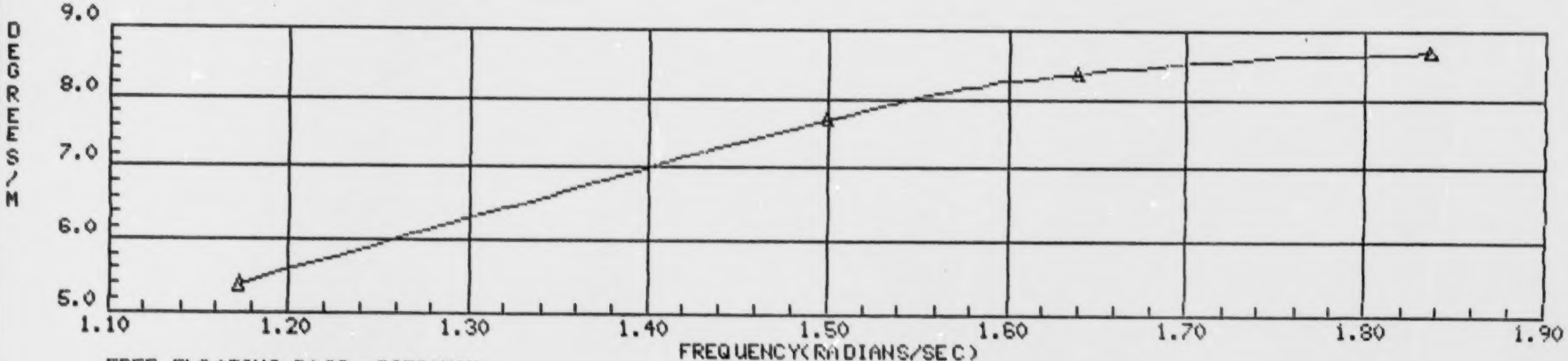


Y*10^{-1} FREE FLOATING RAOS -HEAVE(Z)

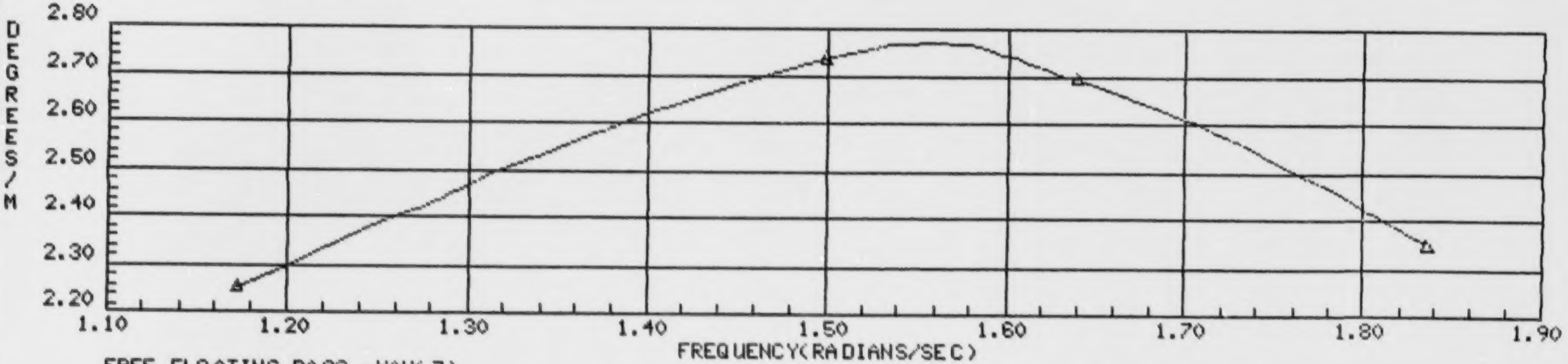
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 3
Progran-AQWA-LINE - EAST COAST VESSEL



FREE FLOATING RAOS -ROLL(X)

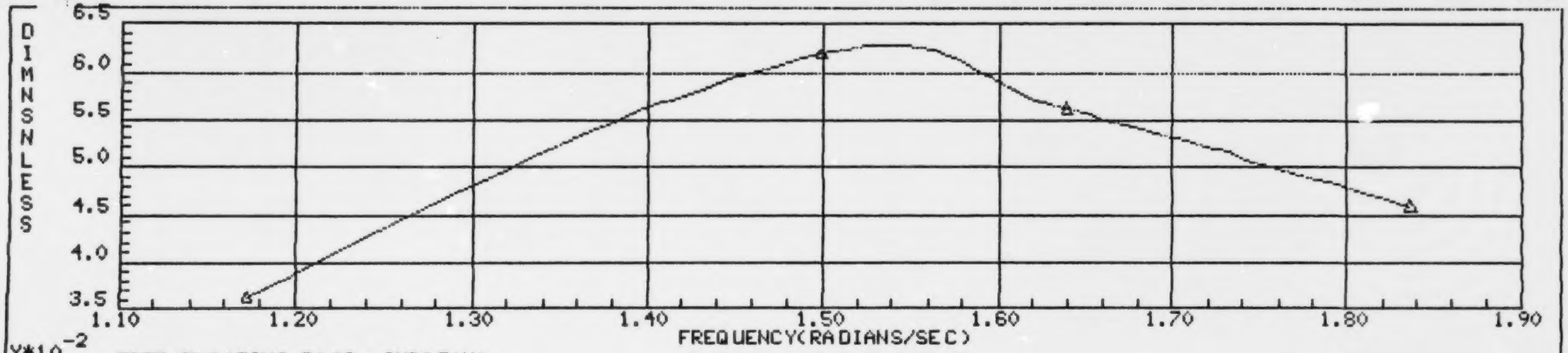


FREE FLOATING RAOS -PITCH(Y)

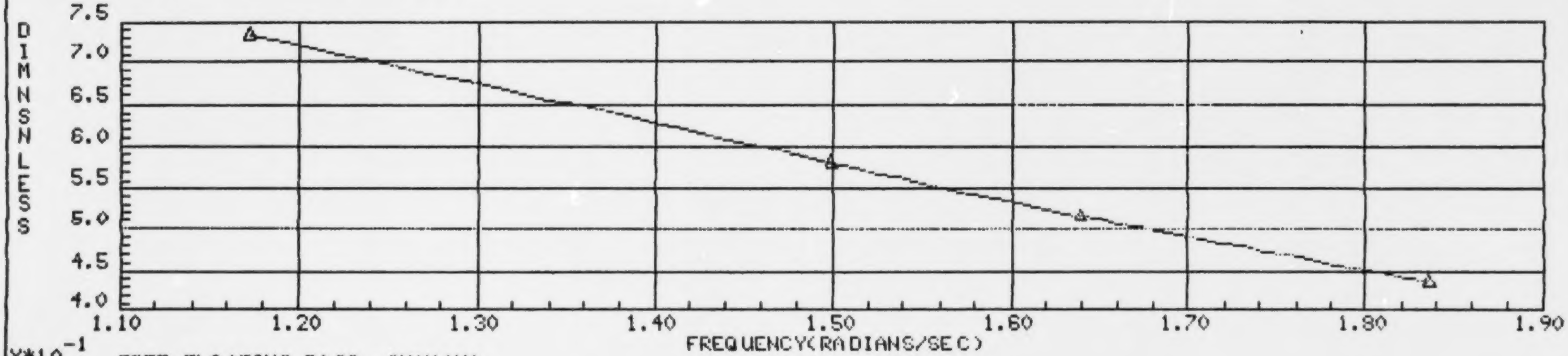


FREE FLOATING RAOS -YAW(Z)

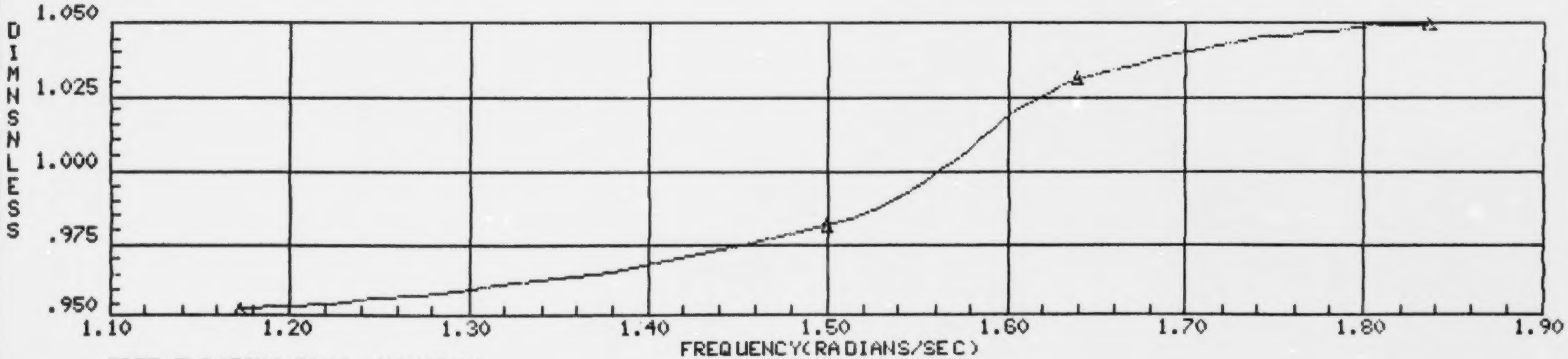
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 3
Program-AQWA-LINE - EAST COAST VESSEL



FREE FLOATING RAOS -SURGE(X)

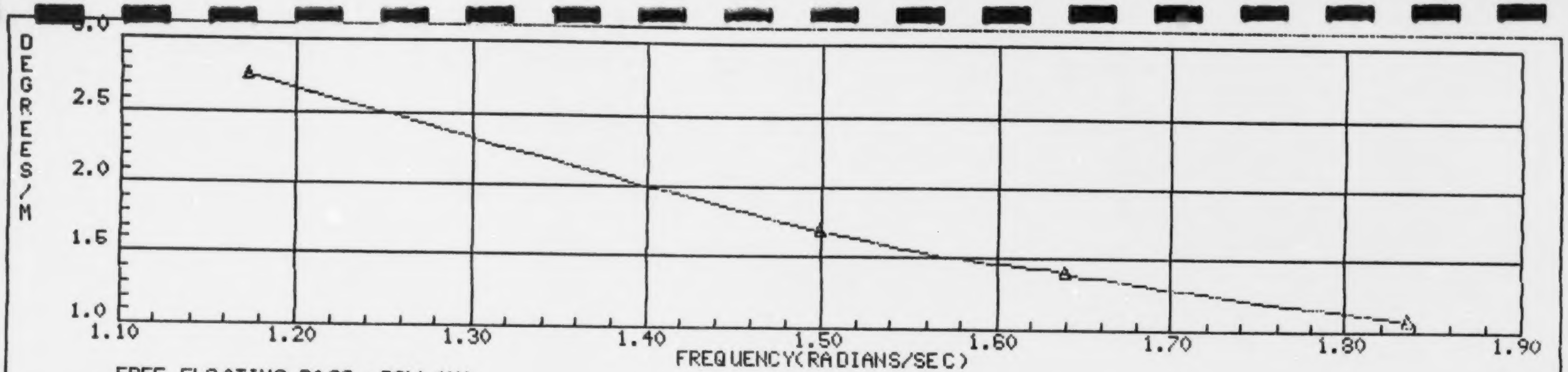


FREE FLOATING RAOS -SWAY(Y)

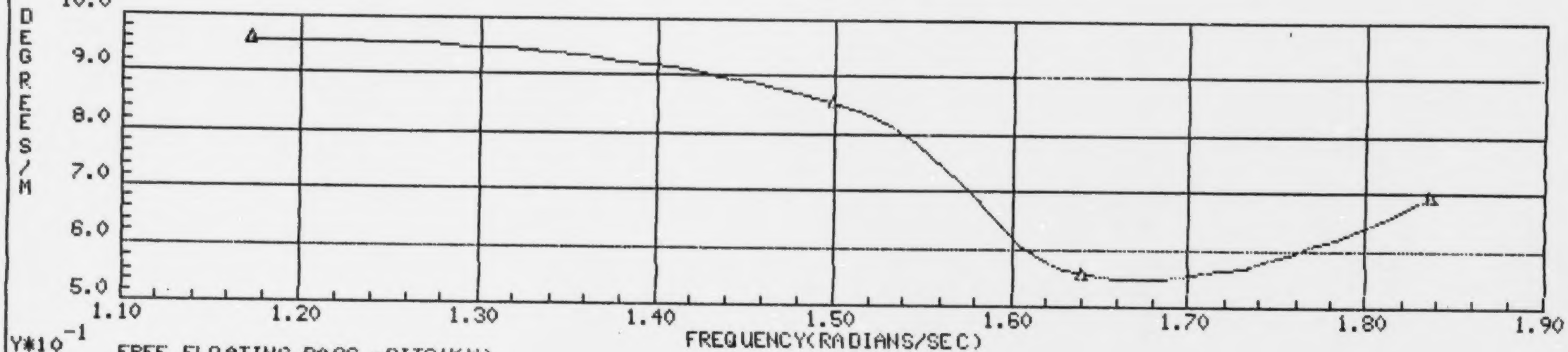


FREE FLOATING RAOS -HEAVE(Z)

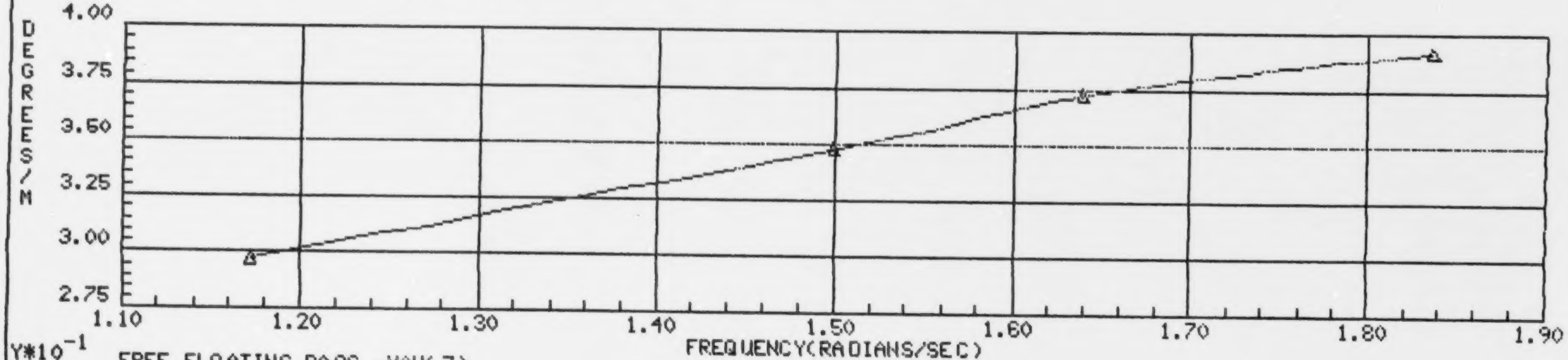
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 4
Progran-AGWA-LINE - EAST COAST VESSEL



FREE FLOATING RAOS -ROLL(X)

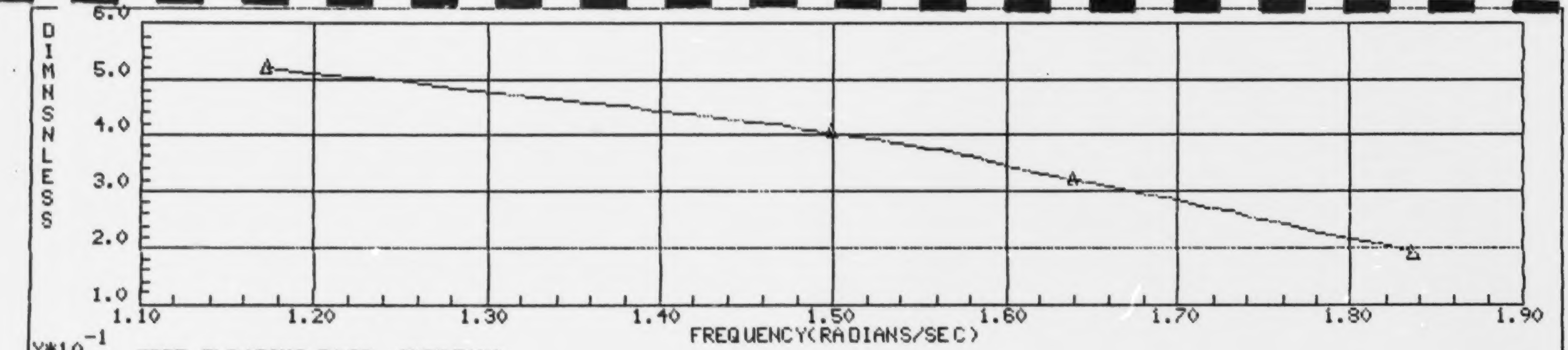


FREE FLOATING RAOS -PITCH(Y)

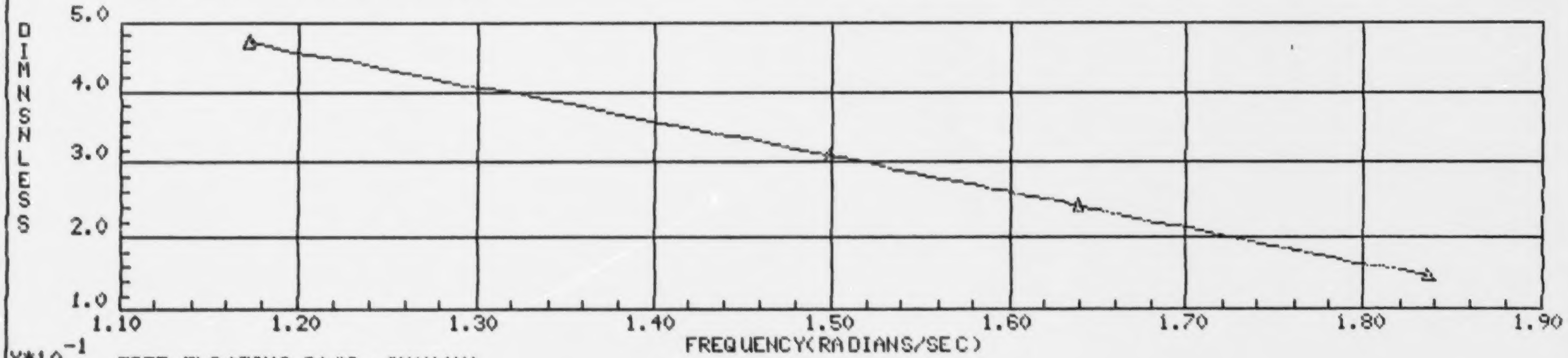


FREE FLOATING RAOS -YAW(Z)

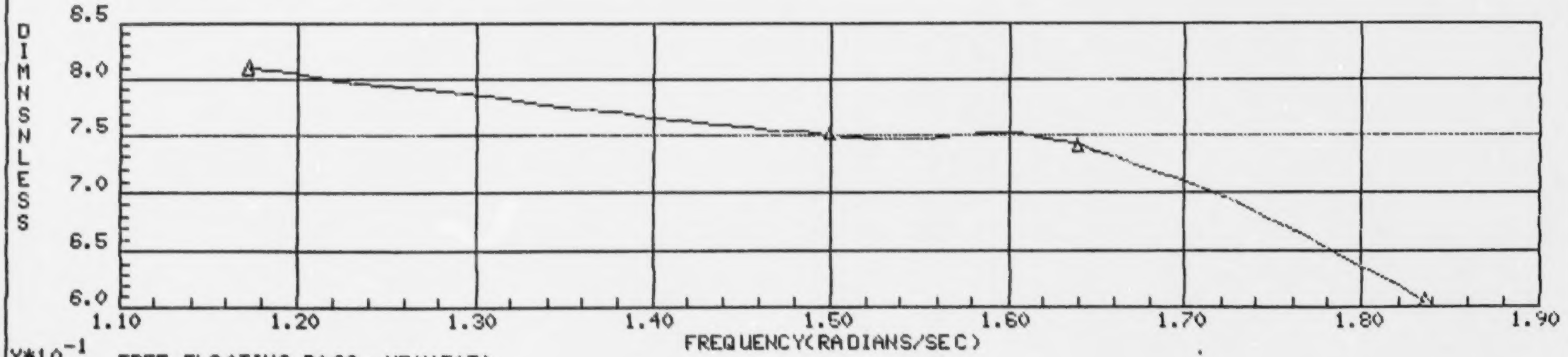
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 4
Program-AQWA-LINE - EAST COAST VESSEL



FREE FLOATING RAOS -SURGE(X)

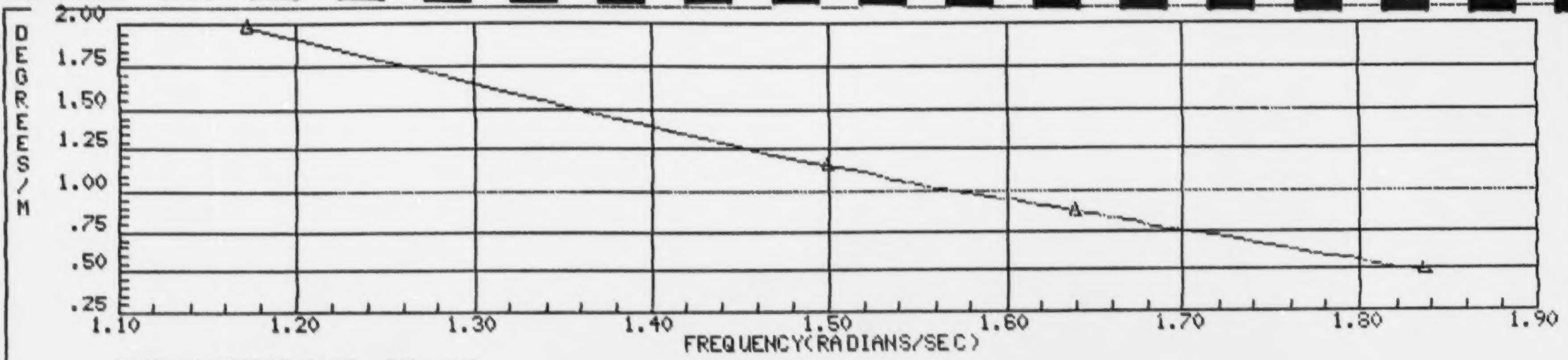


FREE FLOATING RAOS -SWAY(Y)

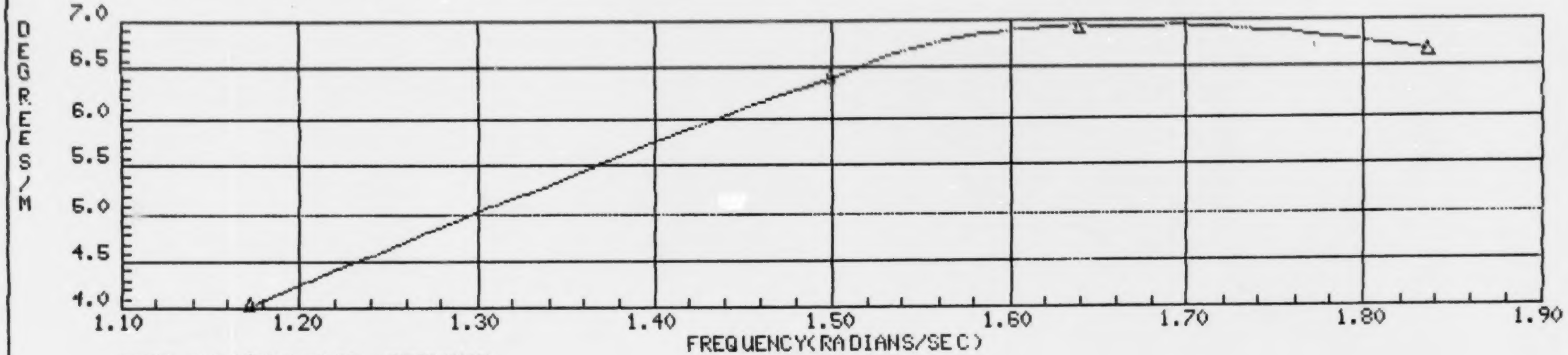


FREE FLOATING RAOS -HEAVE(Z)

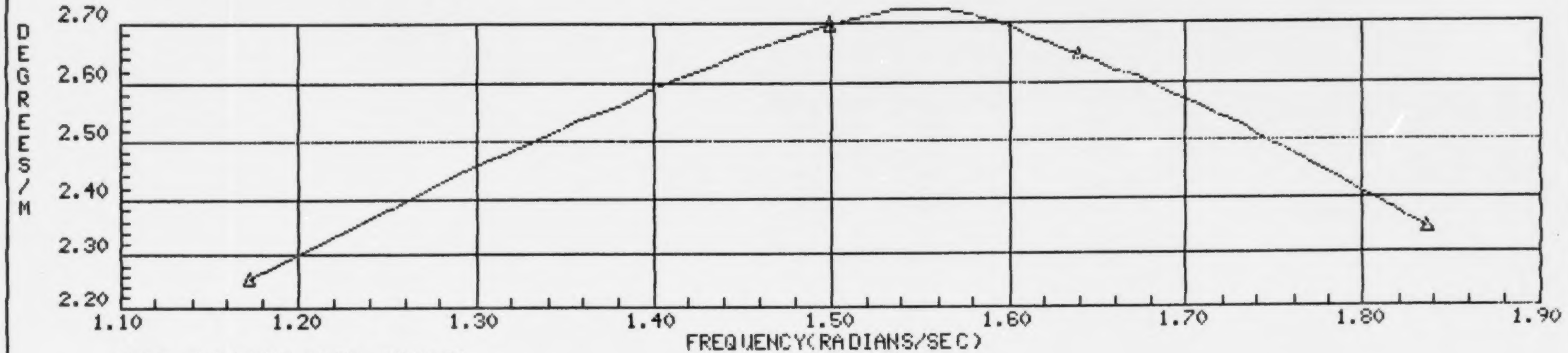
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 5
Progran-AQWA-LINE - EAST COAST VESSEL



FREE FLOATING RAOS -ROLL(X)

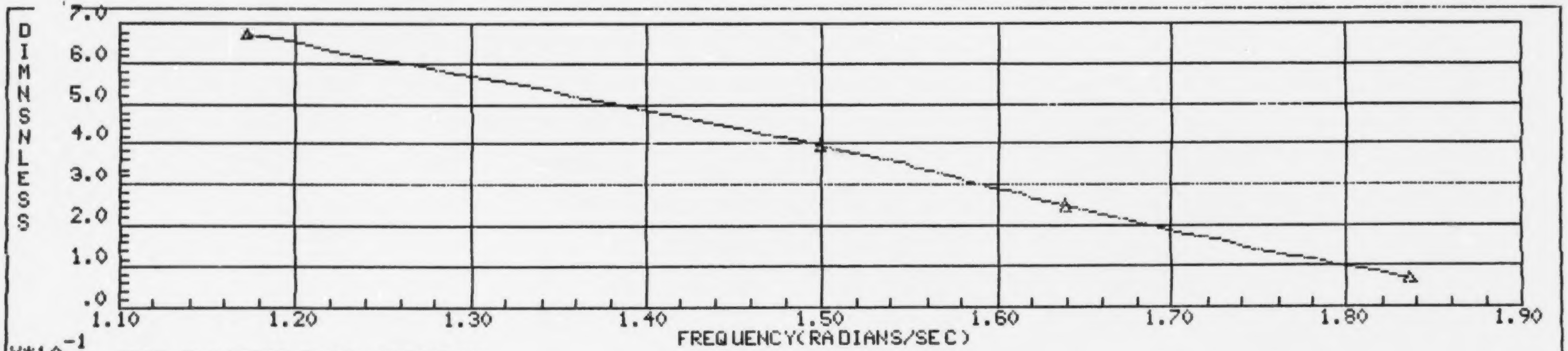


FREE FLOATING RAOS -PITCH(Y)

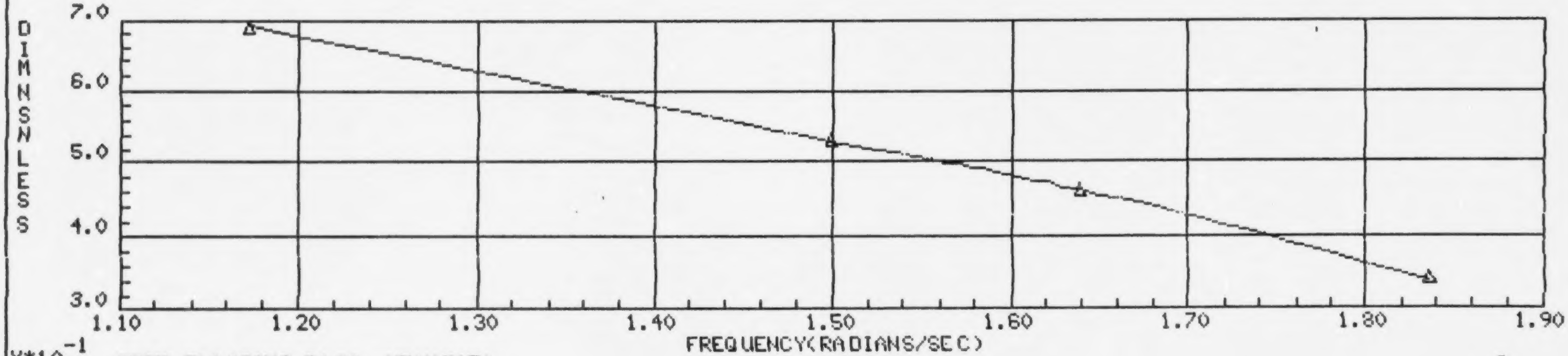


FREE FLOATING RAOS -YAW(Z)

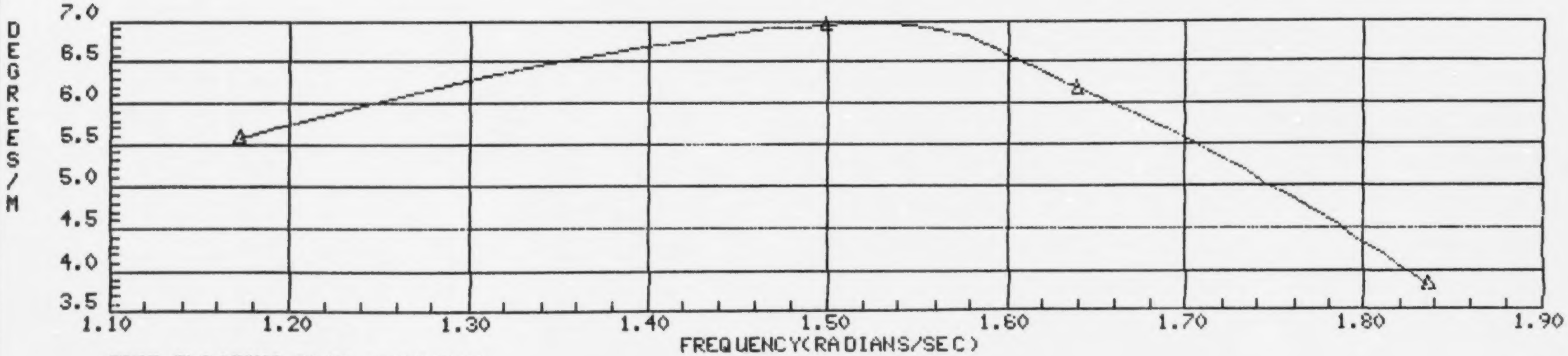
STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 5
Program-AQWA-LINE - EAST COAST VESSEL



FREE FLOATING RAOS -SURGE(X)



FREE FLOATING RAOS -HEAVE(Z)



FREE FLOATING RAOS -PITCH(Y)

STRUCTURE 1 FREQUENCY VARYING PARAMETERS-DIRECTION NUMBER 6
Progran-AQWA-LINE - EAST COAST VESSEL

APPENDIX D

RESULTS AT SUBHARMONIC RESONANCE

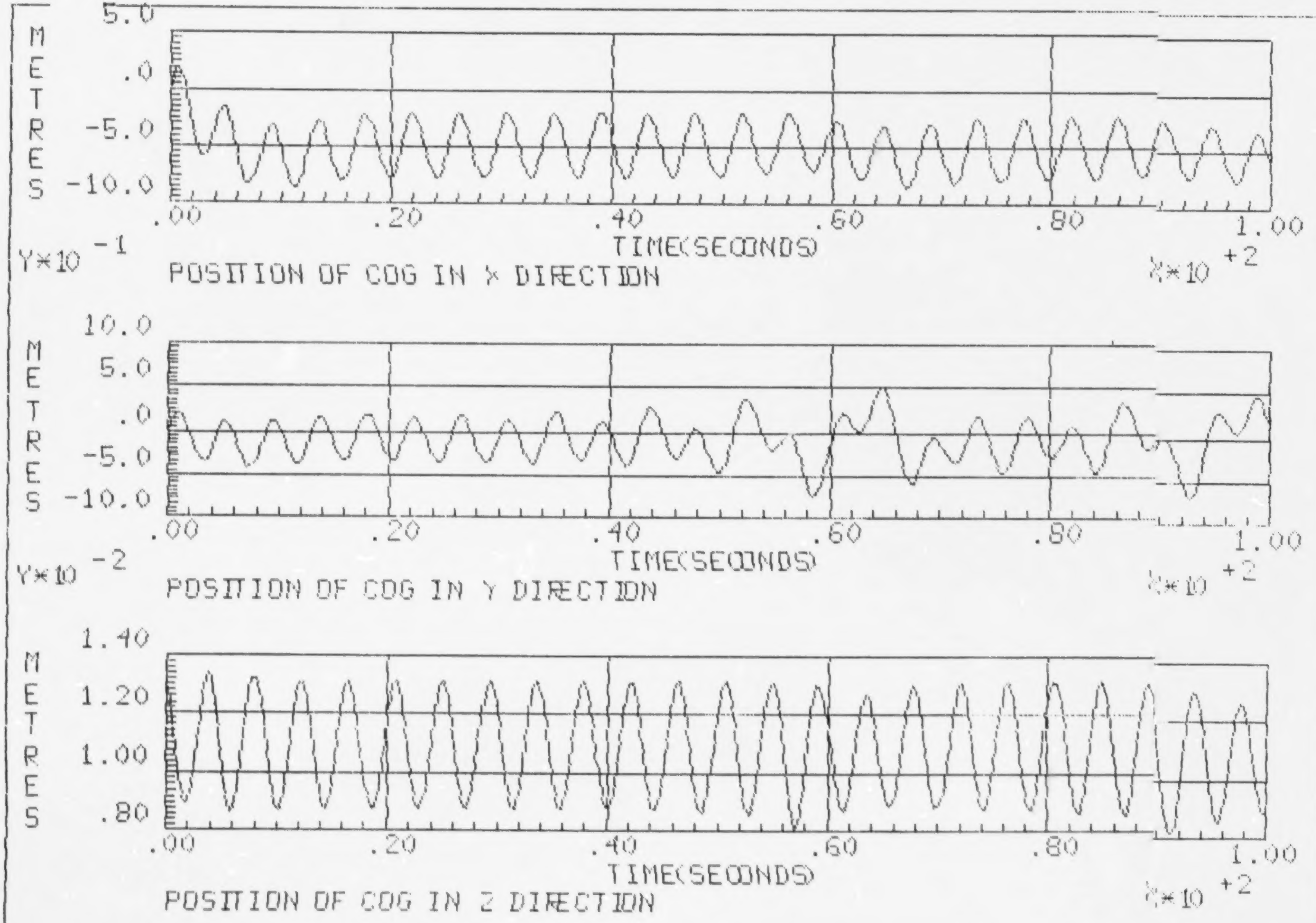
ENCOUNTERING FREQUENCY

HARD CHINE VESSEL

Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
WM1	5.0	1/12	0.4
WM2	30.0	1/12	0.4
WM3	50.0	1/12	0.4
WM4	5.0	1/7	0.4
WM5	30.0	1/7	0.4
WM6	50.0	1/7	0.4
WM4-5%	5.0	1/7	5.0

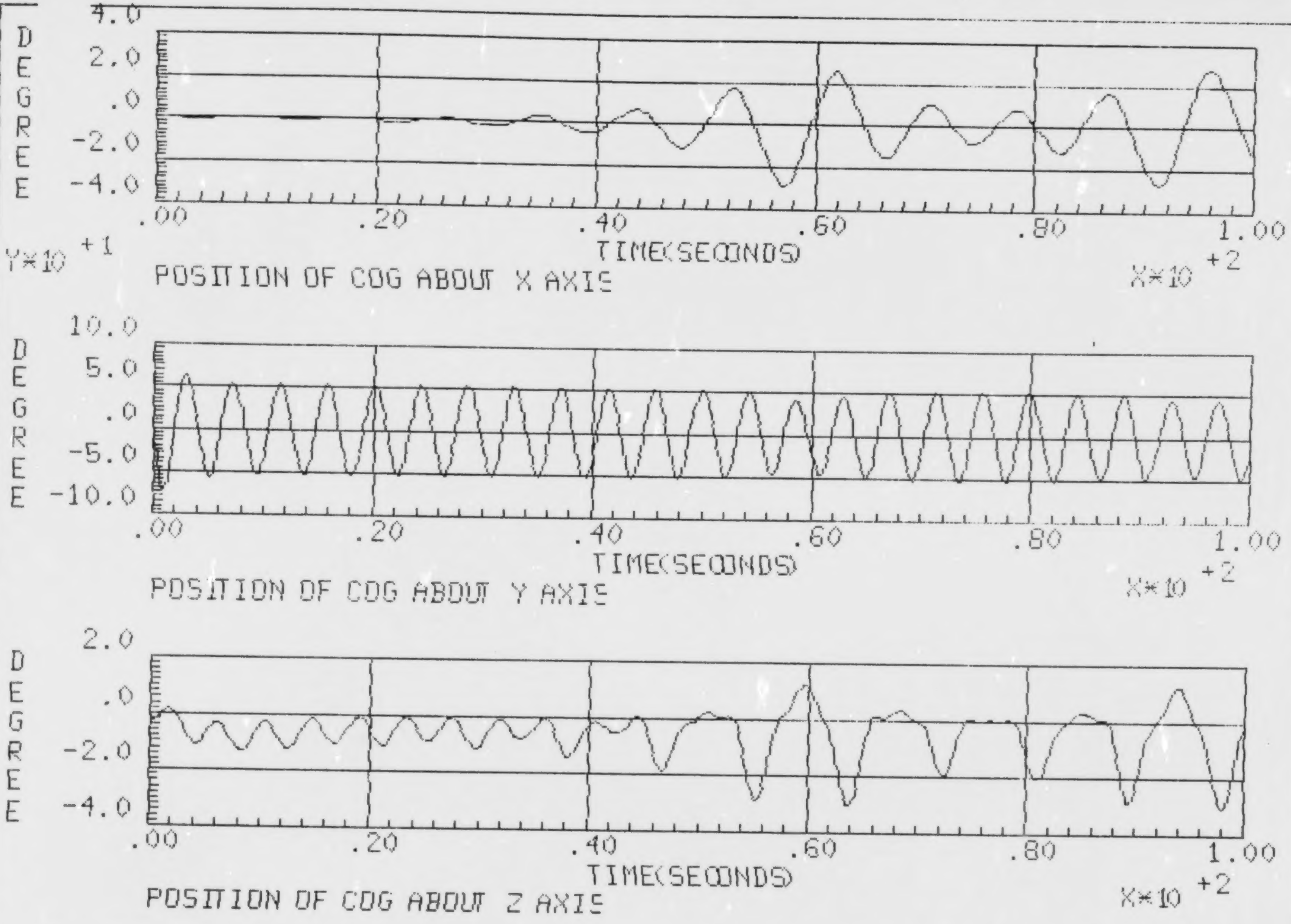
ROUND BILGE VESSEL

Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
EM4	5.0	1/7	0.4
EM4-5%	5.0	1/7	5.0



HARD CHINE VESSEL CASE NO. WM1

Te=0.5 Tr, Dir.=5 °, H=1.91 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



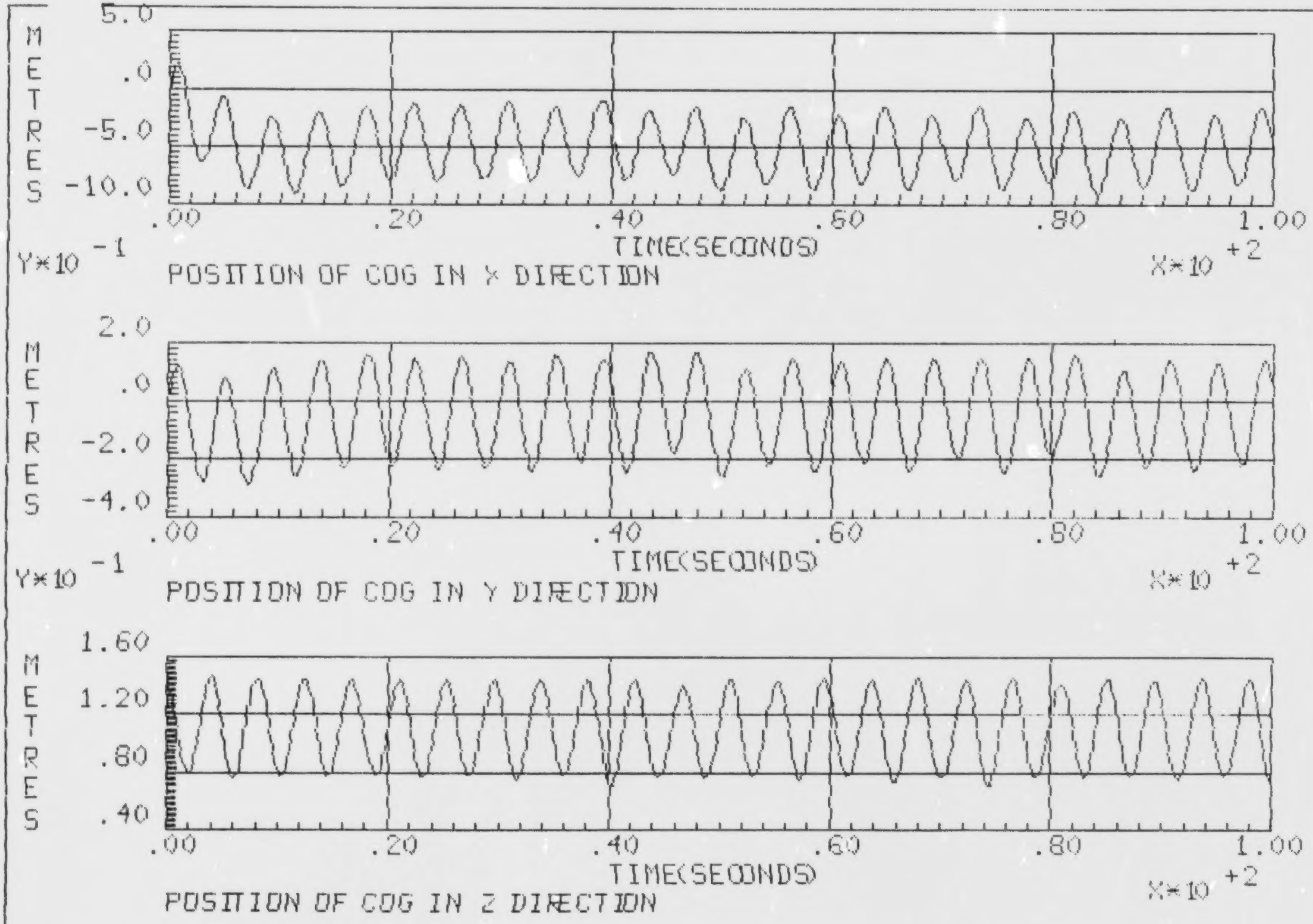
POSITION OF COG ABOUT X AXIS

POSITION OF COG ABOUT Y AXIS

POSITION OF COG ABOUT Z AXIS

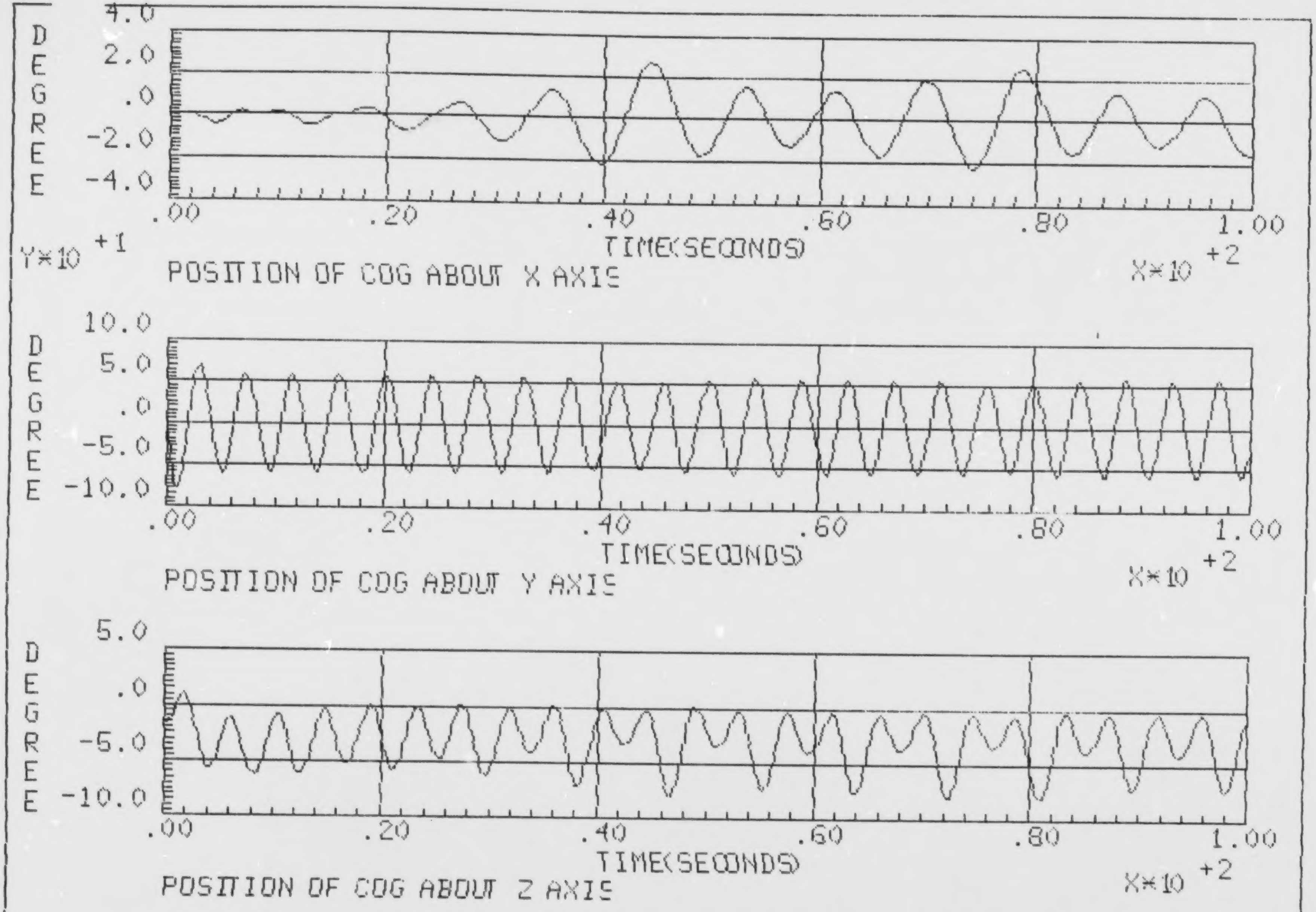
HARD CHINE VESSEL CASE NO. WM1

Te=0.5 Tr, Dir.=5 °, H=1.91 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



HARD CHINE VESSEL CASE NO. WM2

Te=0.5 Tr, Dir.=30 °, H=1.91 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



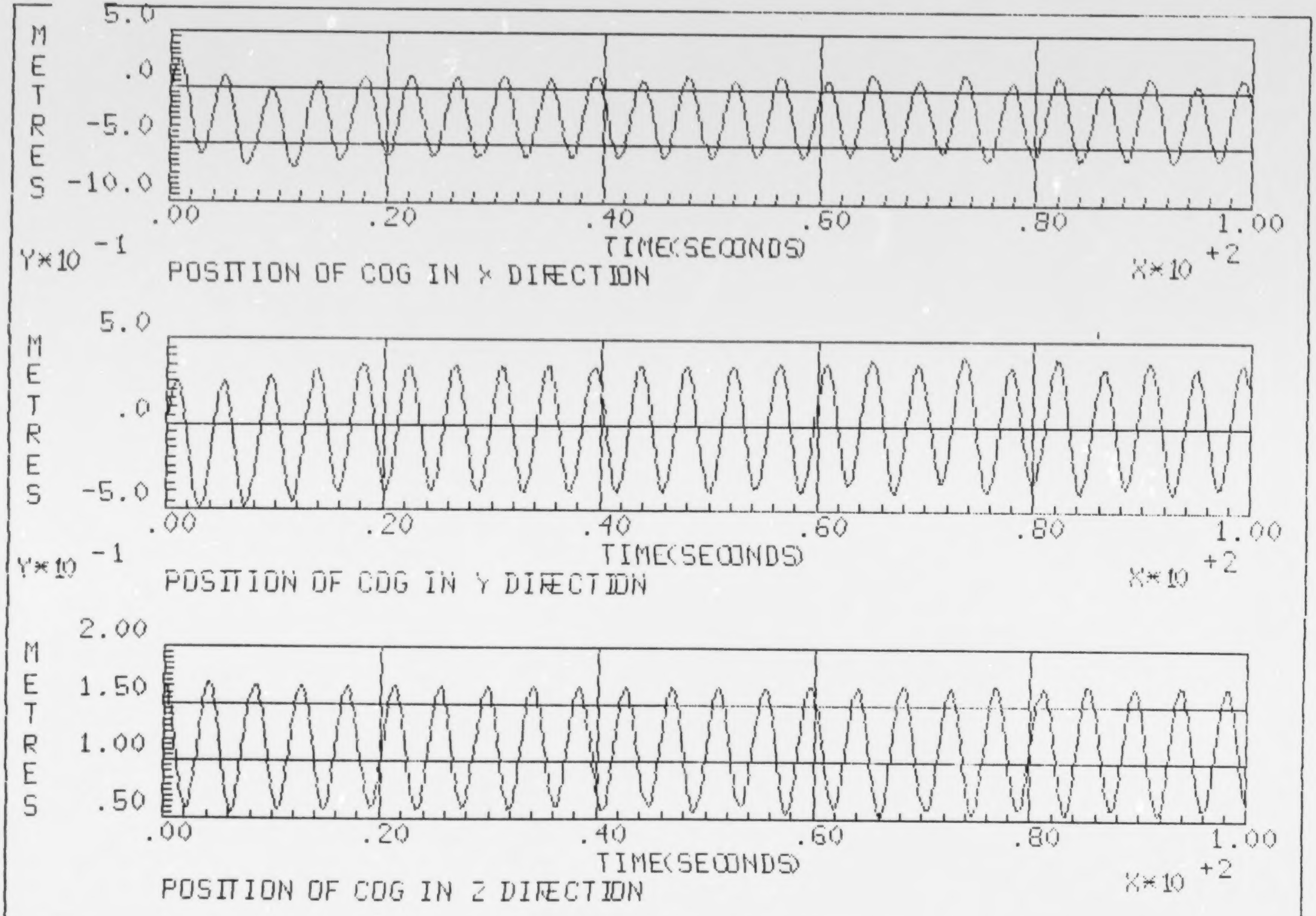
POSITION OF COG ABOUT X AXIS

POSITION OF COG ABOUT Y AXIS

POSITION OF COG ABOUT Z AXIS

HARD CHINE VESSEL CASE NO. WM2

Te=0.5 Tr, Dir.=30 °, H=1.91 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



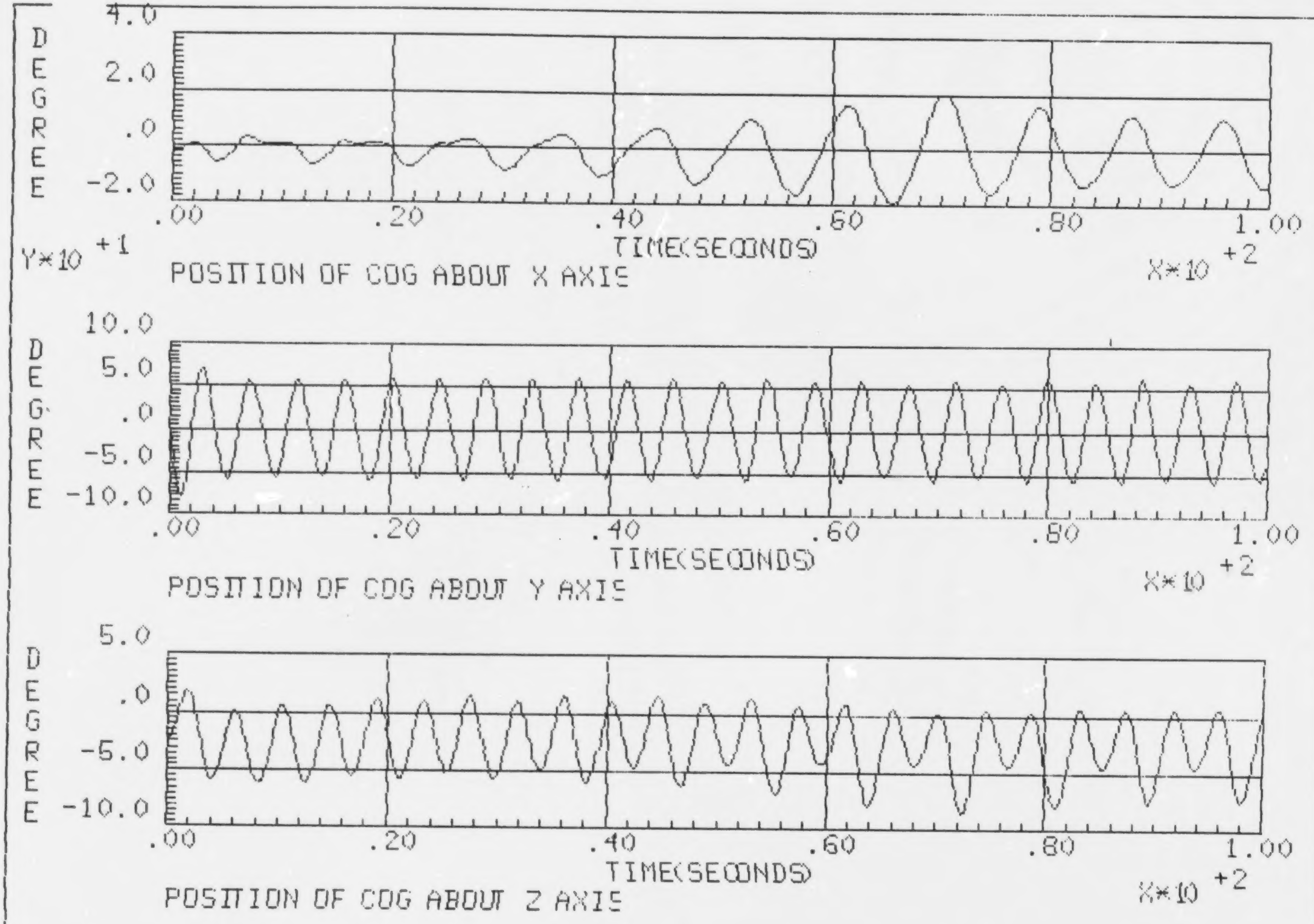
POSITION OF COG IN X DIRECTION

POSITION OF COG IN Y DIRECTION

POSITION OF COG IN Z DIRECTION

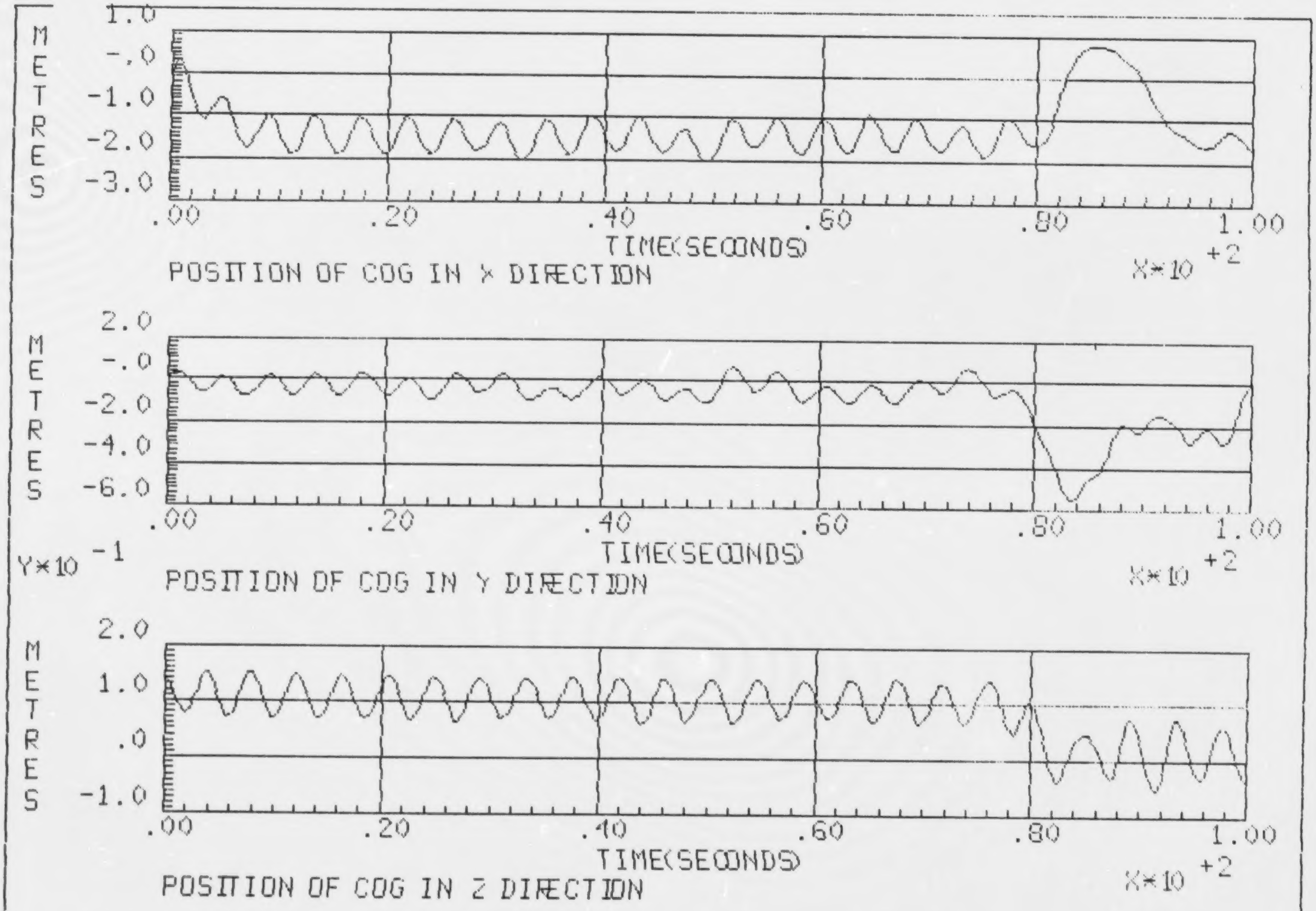
HARD CHINE VESSEL CASE NO. WM3

Te=0.5 Tr, Dir.=50 °, H=1.91 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



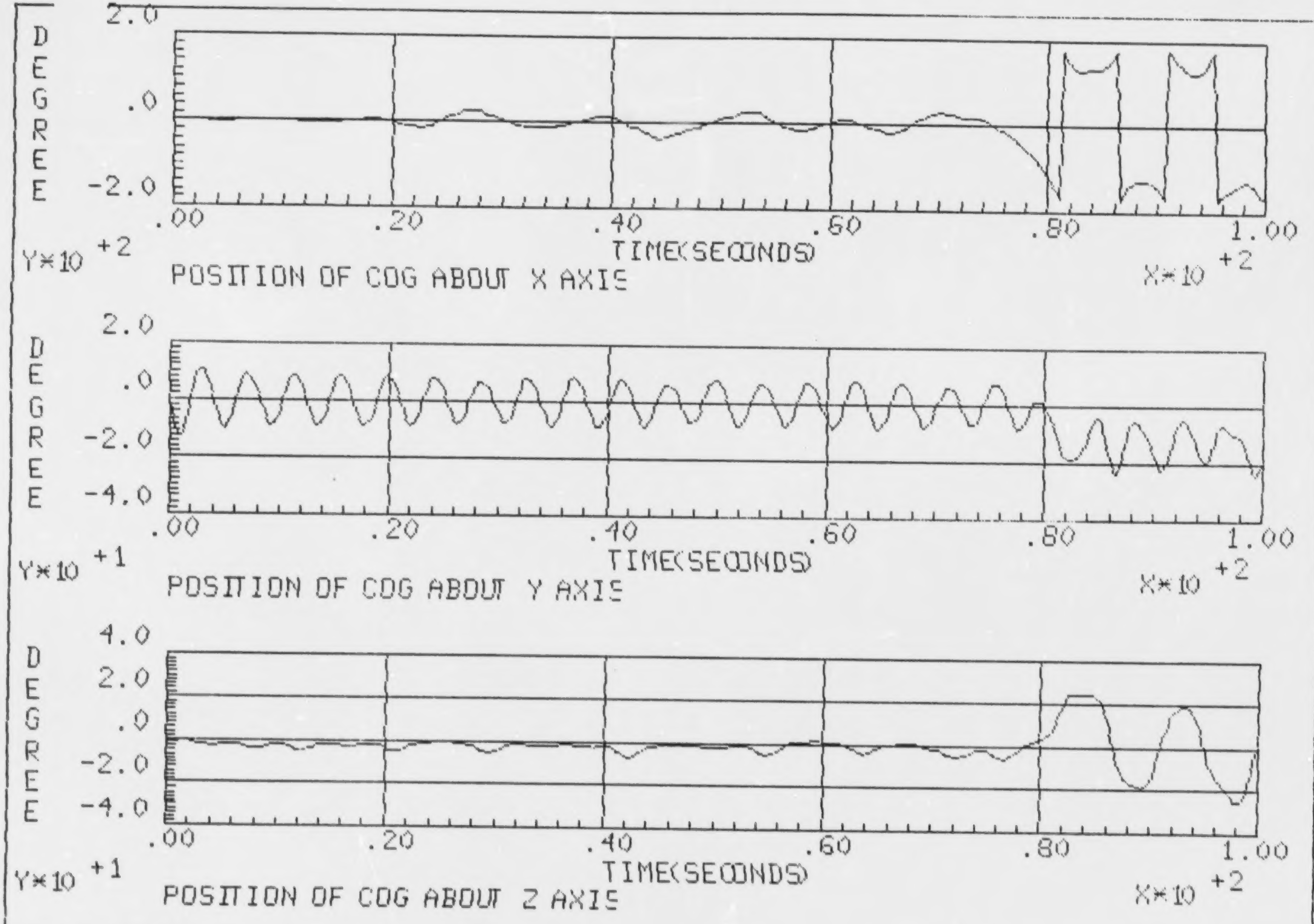
HARD CHINE VESSEL CASE NO. WM3

Te=0.5 Tr, Dir.=50 °, H=1.91 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



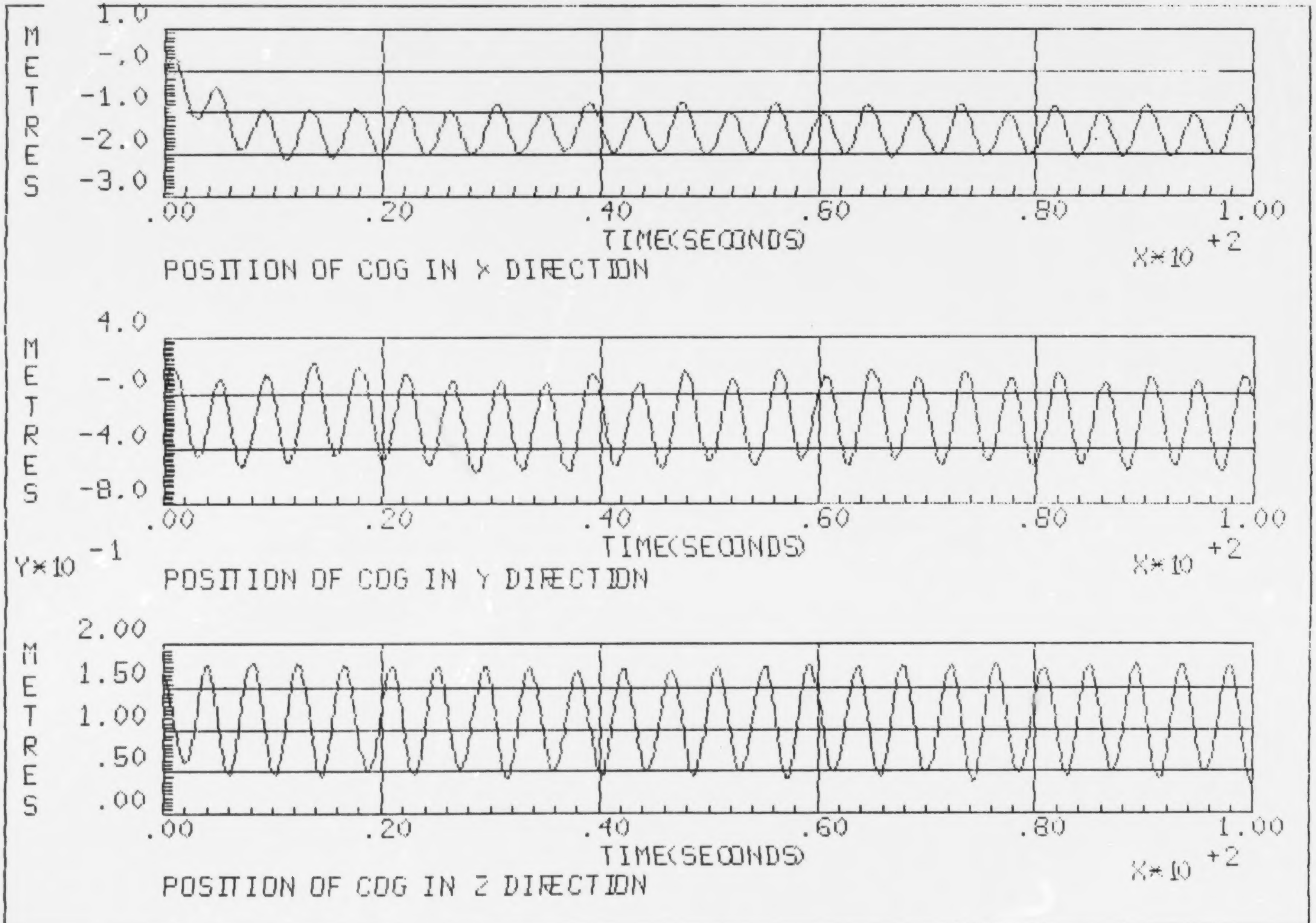
HARD CHINE VESSEL CASE NO. WM4

Te=0.5 Tr, Dir.=5 °, H=3.24 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



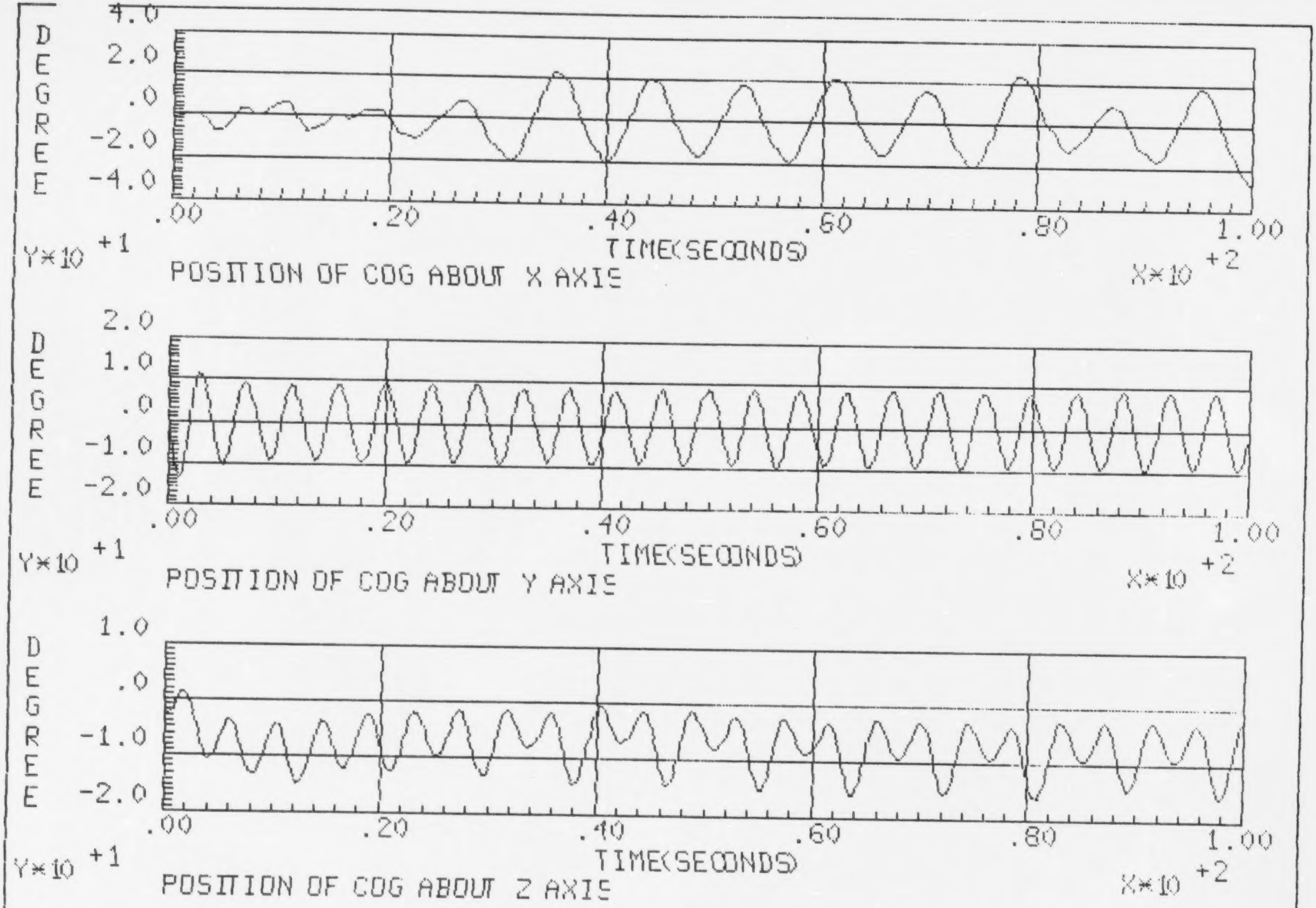
HARD CHINE VESSEL CASE NO. WM4

Te=0.5 Tr, Dir.=5 °, H=3.24 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



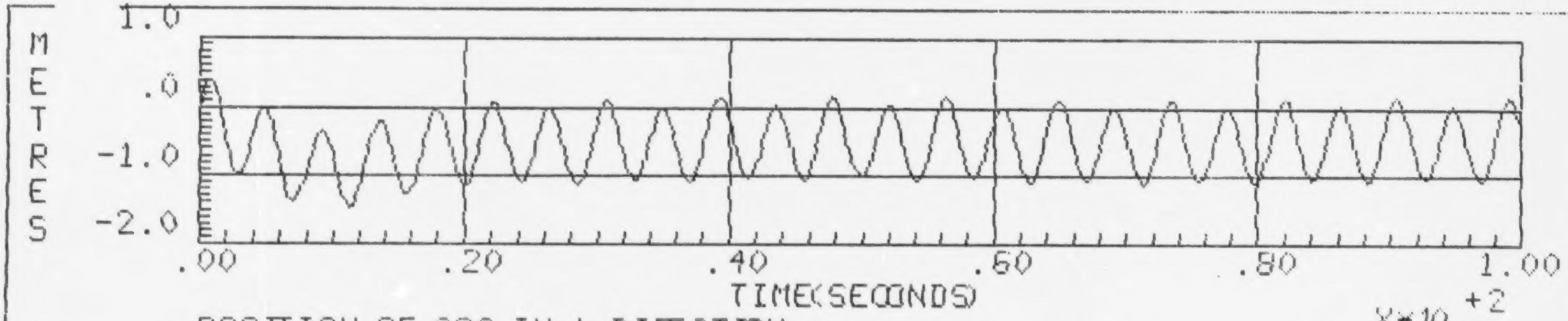
HARD CHINE VESSEL. CASE NO. WM5

Te=0.5 Tr, Dir.=30 °, H=3.24 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1

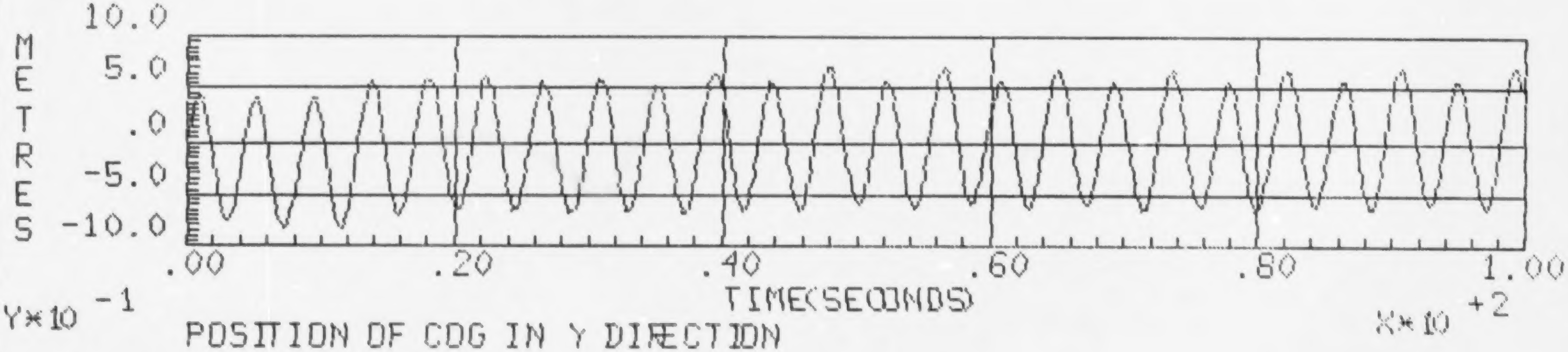


HARD CHINE VESSEL CASE NO. WM5

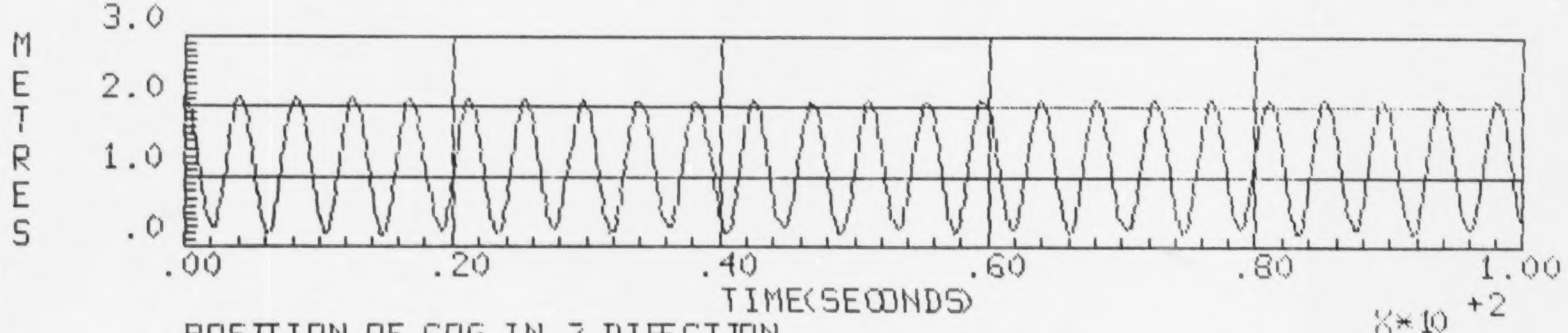
Te=0.5 Tr, Dir.=30 °, H=3.24 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



POSITION OF COG IN X DIRECTION



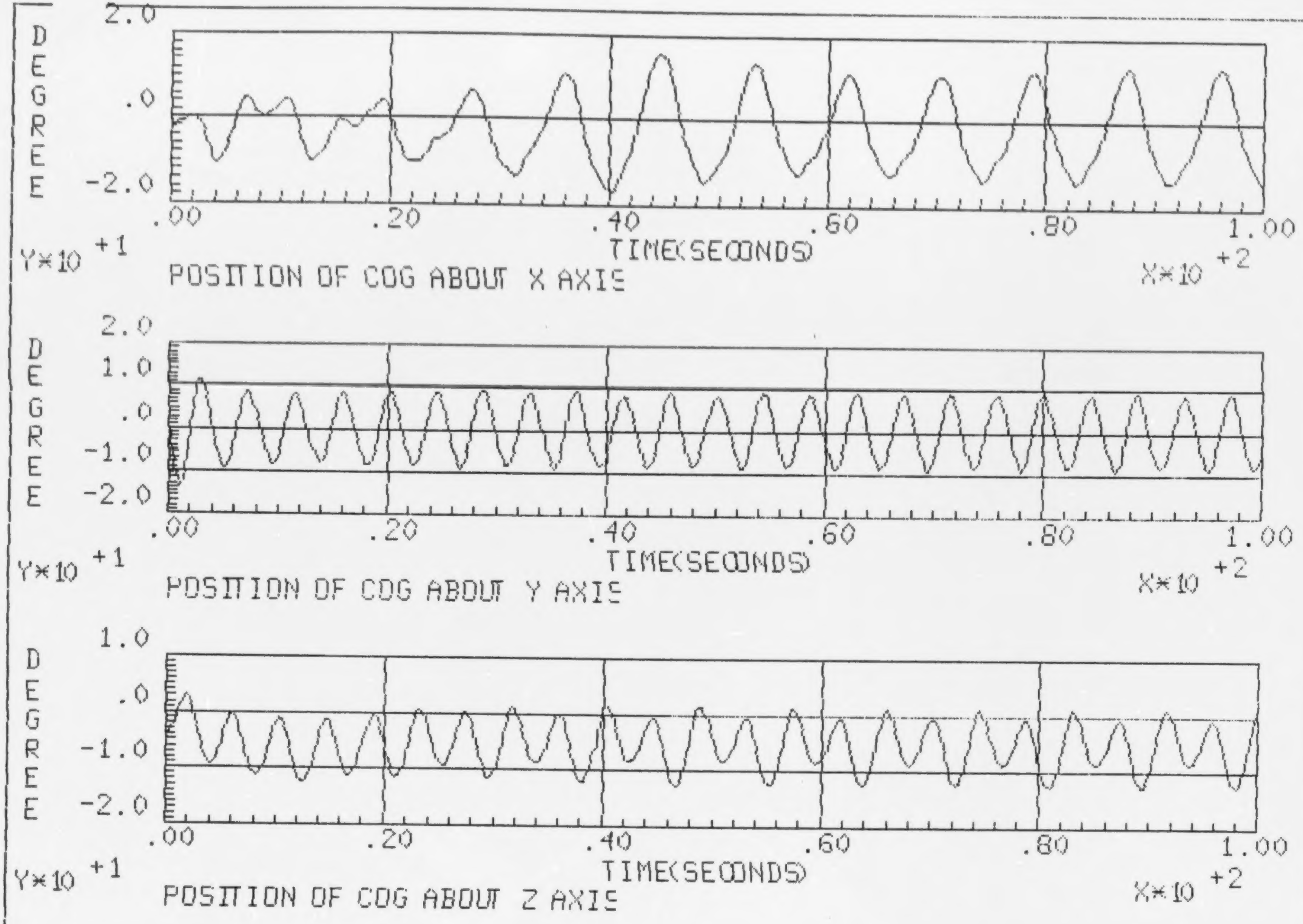
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

HARD CHINE VESSEL CASE NO. WM6

Te=0.5 Tr, Dir.=50 °, H=3.24 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



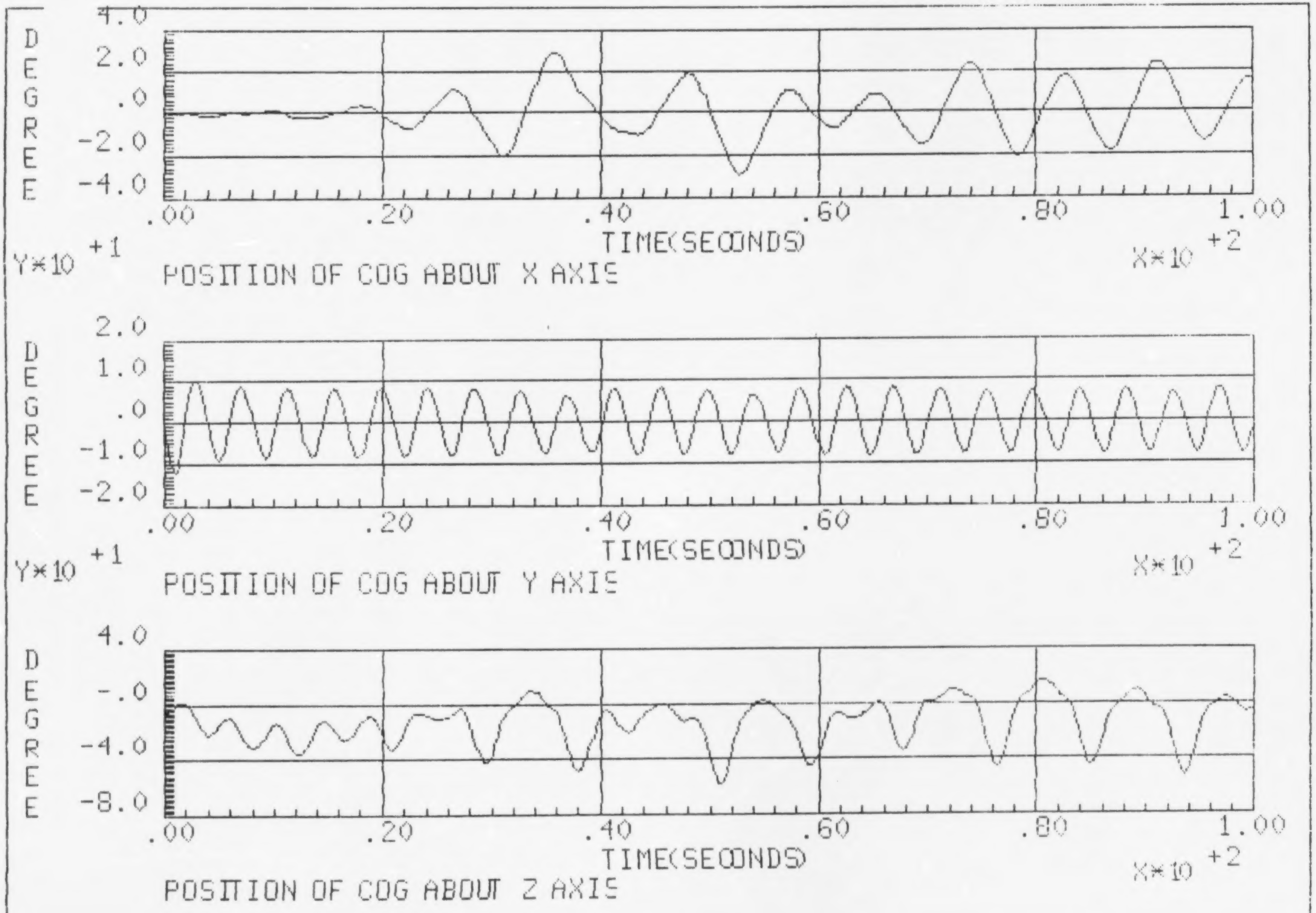
POSITION OF COG ABOUT X AXIS

POSITION OF COG ABOUT Y AXIS

POSITION OF COG ABOUT Z AXIS

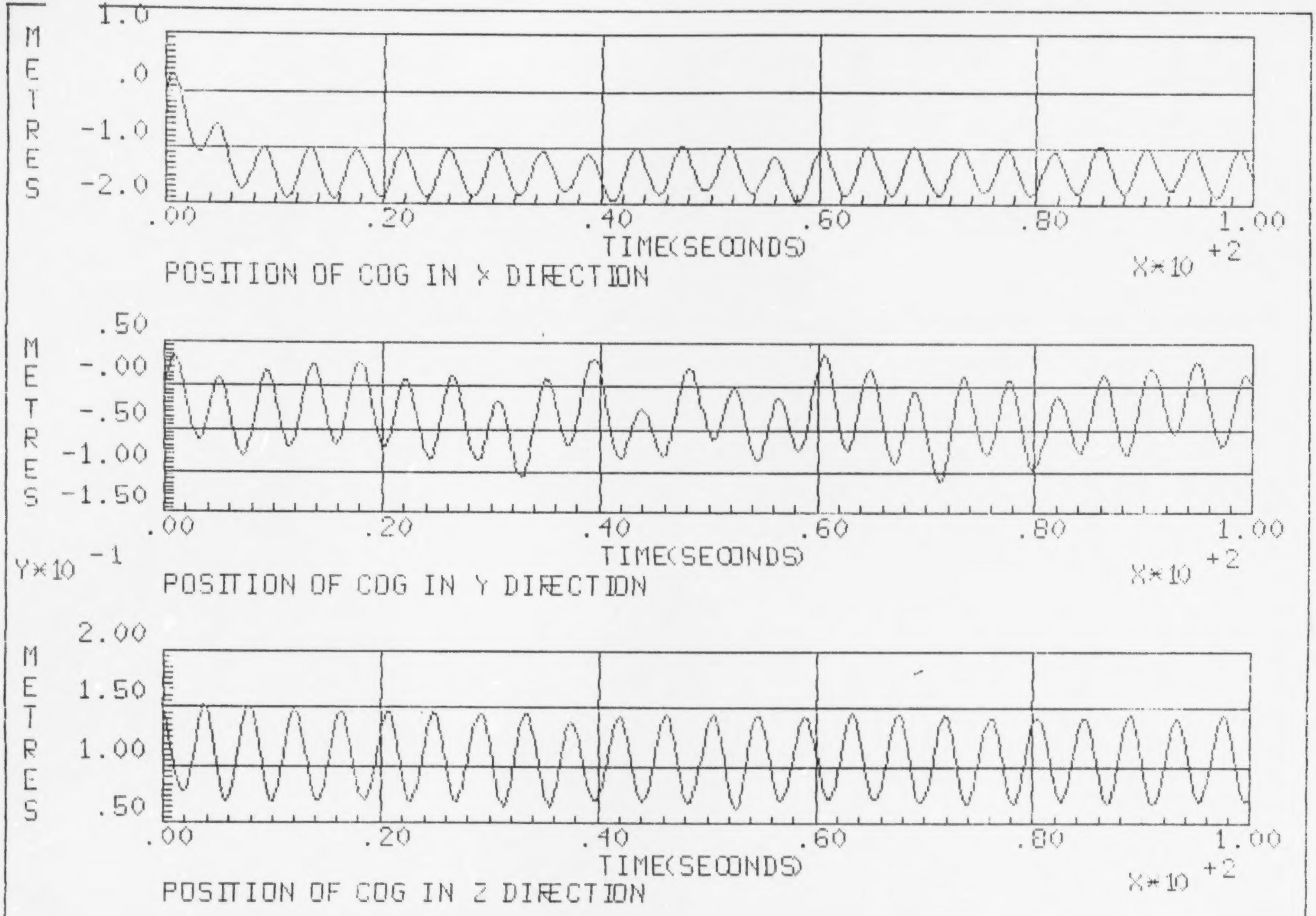
HARD CHINE VESSEL CASE NO. WM6

Te=0.5 Tr, Dir.=50 °, H=3.24 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



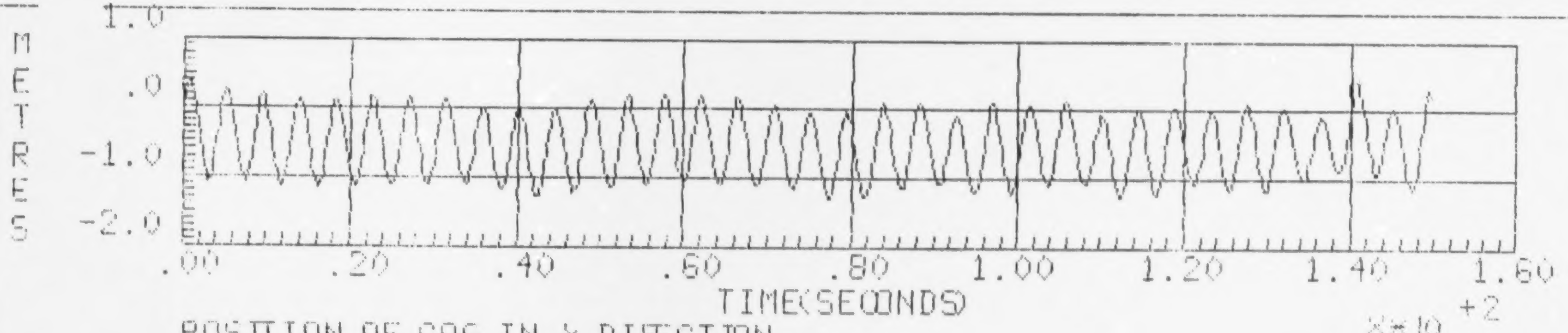
HARD CHINE VESSEL CASE NO. WM4-5%

Te=0.5 Tr, Dir.=5 °, H=1.91 m, Diff.<>0, D_{44}=5.0 % CRIT., Servo 1

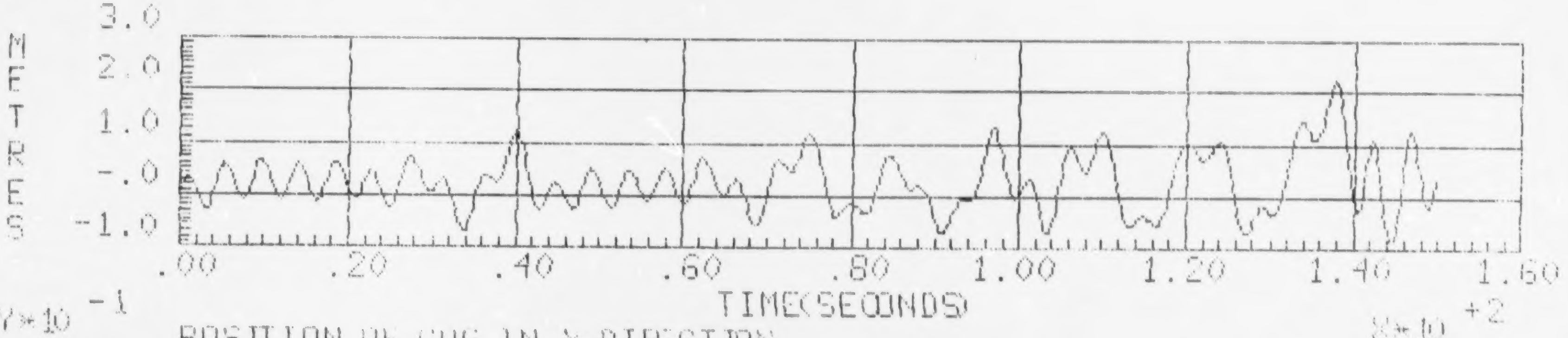


HARD CHINE VESSEL CASE NO. WM4-5%

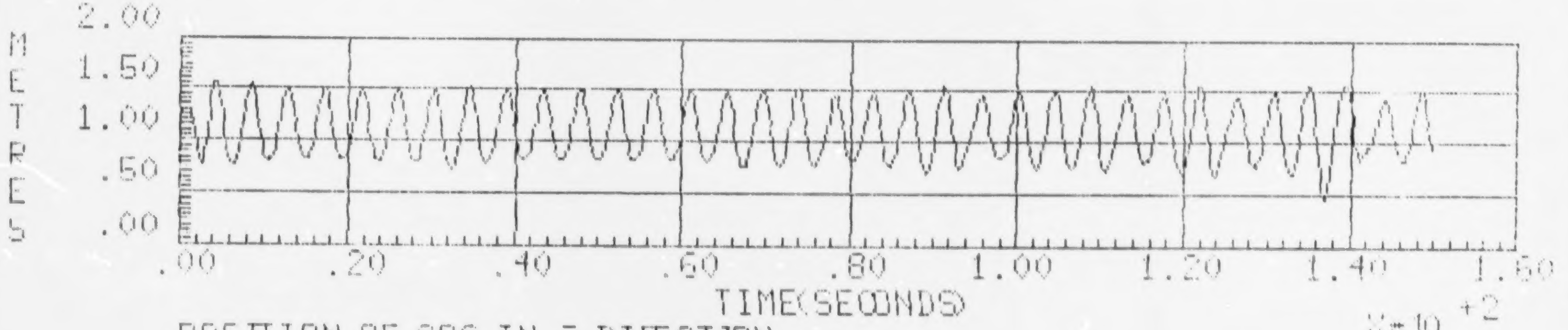
Te=0.5 Tr, Dir.=5 °, H=1.91 m, Diff.<>0, D_{44}=5.0 % CRIT., Servo 1



POSITION OF COG IN X DIRECTION



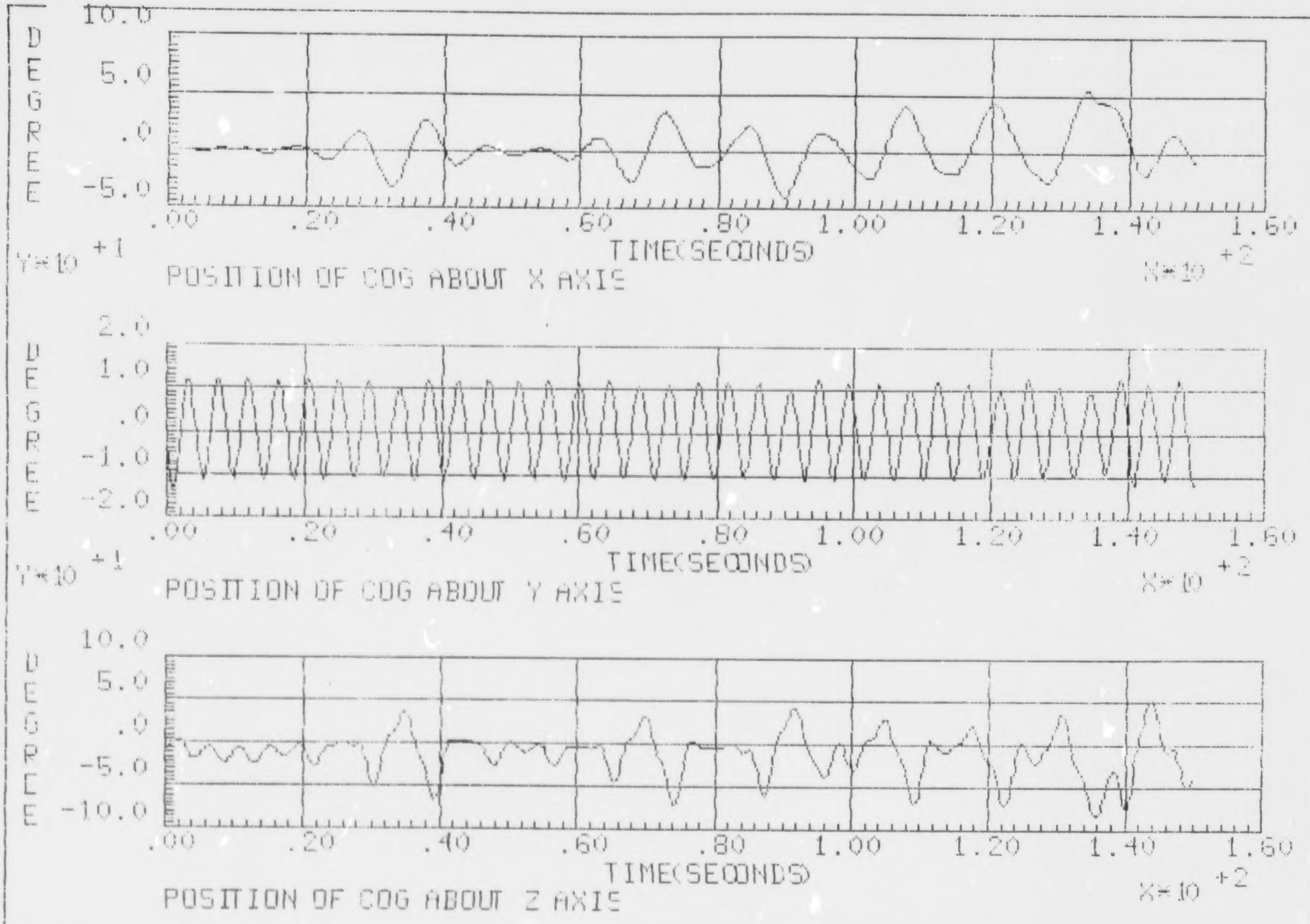
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

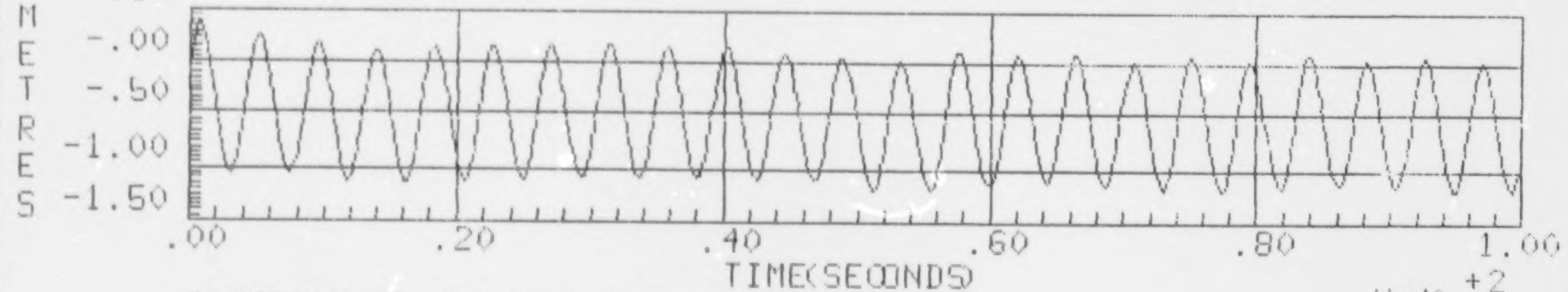
ROUND BILGE VESSEL CASE NO. EM4

Te=0.5 Tr, Dir.=5 °, H=3.24 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1

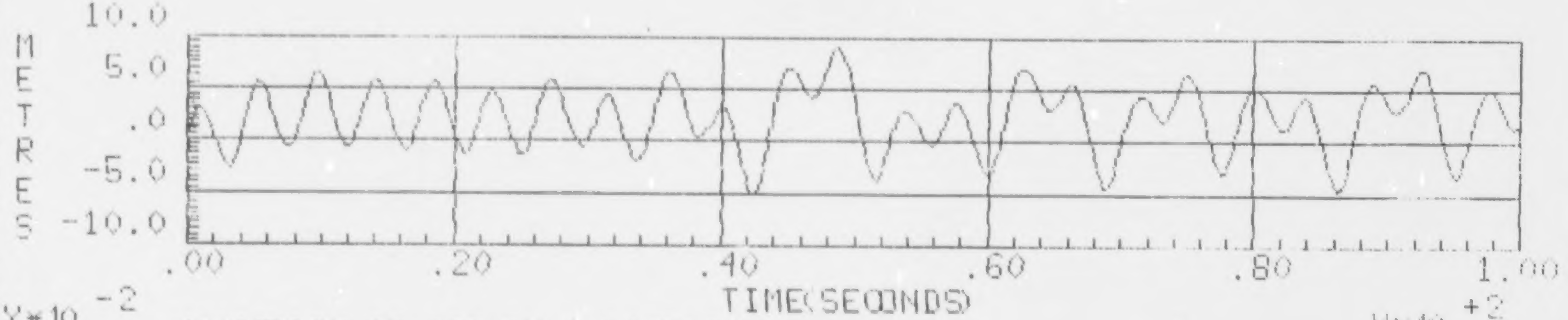


ROUND BILGE VESSEL CASE NO. EM4

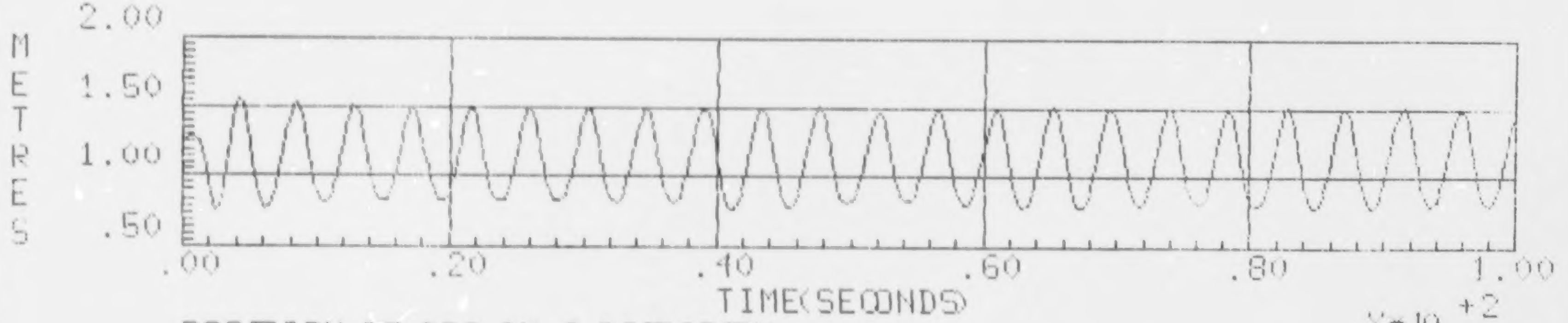
Te=0.5 Tr, Dir.=5 °, H=3.24 m, Diff.<>0, D_{44}=0.4 % CRIT., Servo 1



POSITION OF COG IN X DIRECTION



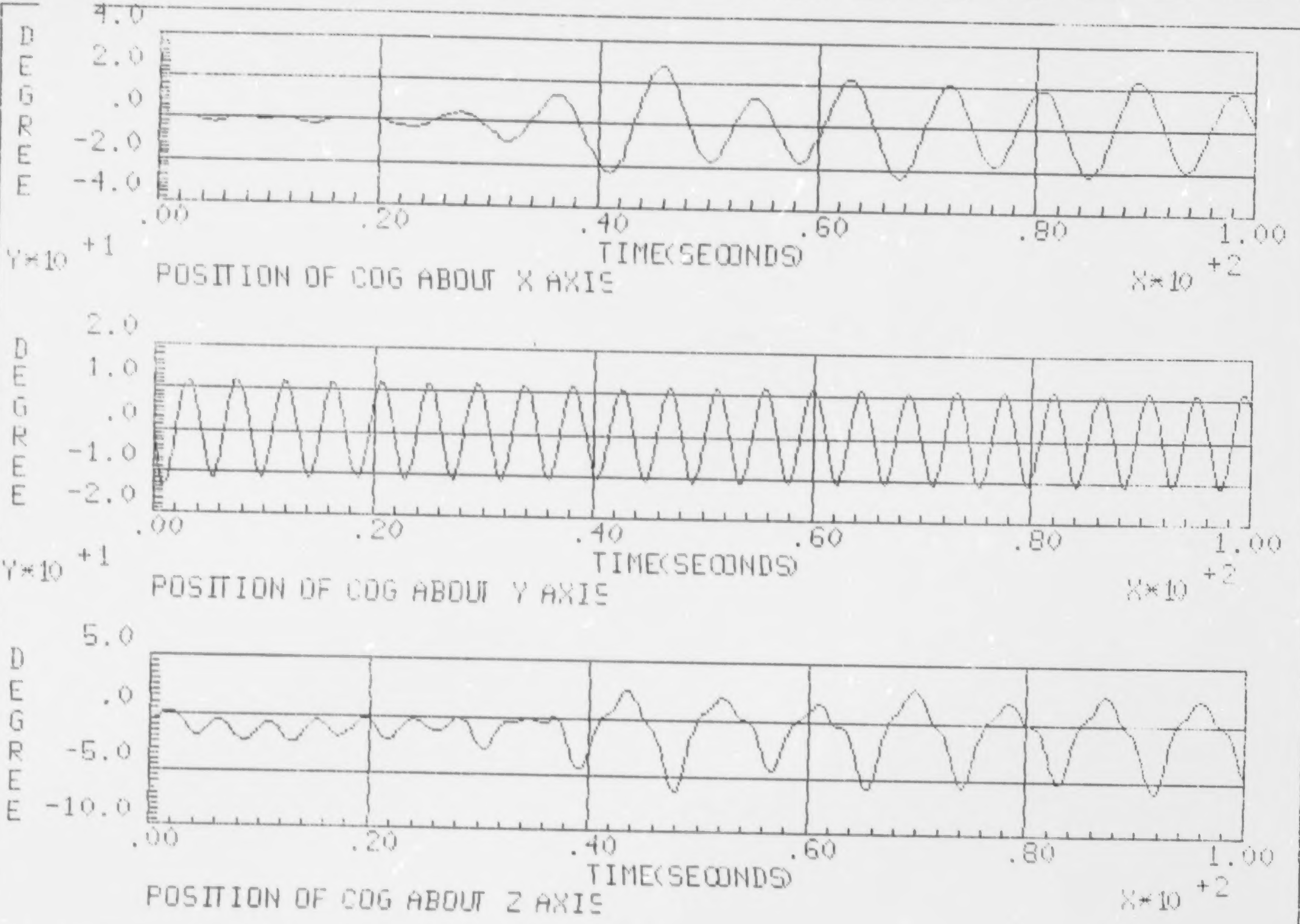
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

ROUND BILGE VESSEL CASE NO. EM4-5%

Te=0.5 Tr, Dir.=5 °, H=3.24 m, Diff.<>0, D_{44}=5.0 % CRIT., Servo 1



ROUND BILGE VESSEL. CASE NO. EM4-5%

Te=0.5 Tr, Dir.=5 °, H=3.24 m, Diff.<>0, D_{44}=5.0 % CRIT., Servo 1

APPENDIX E

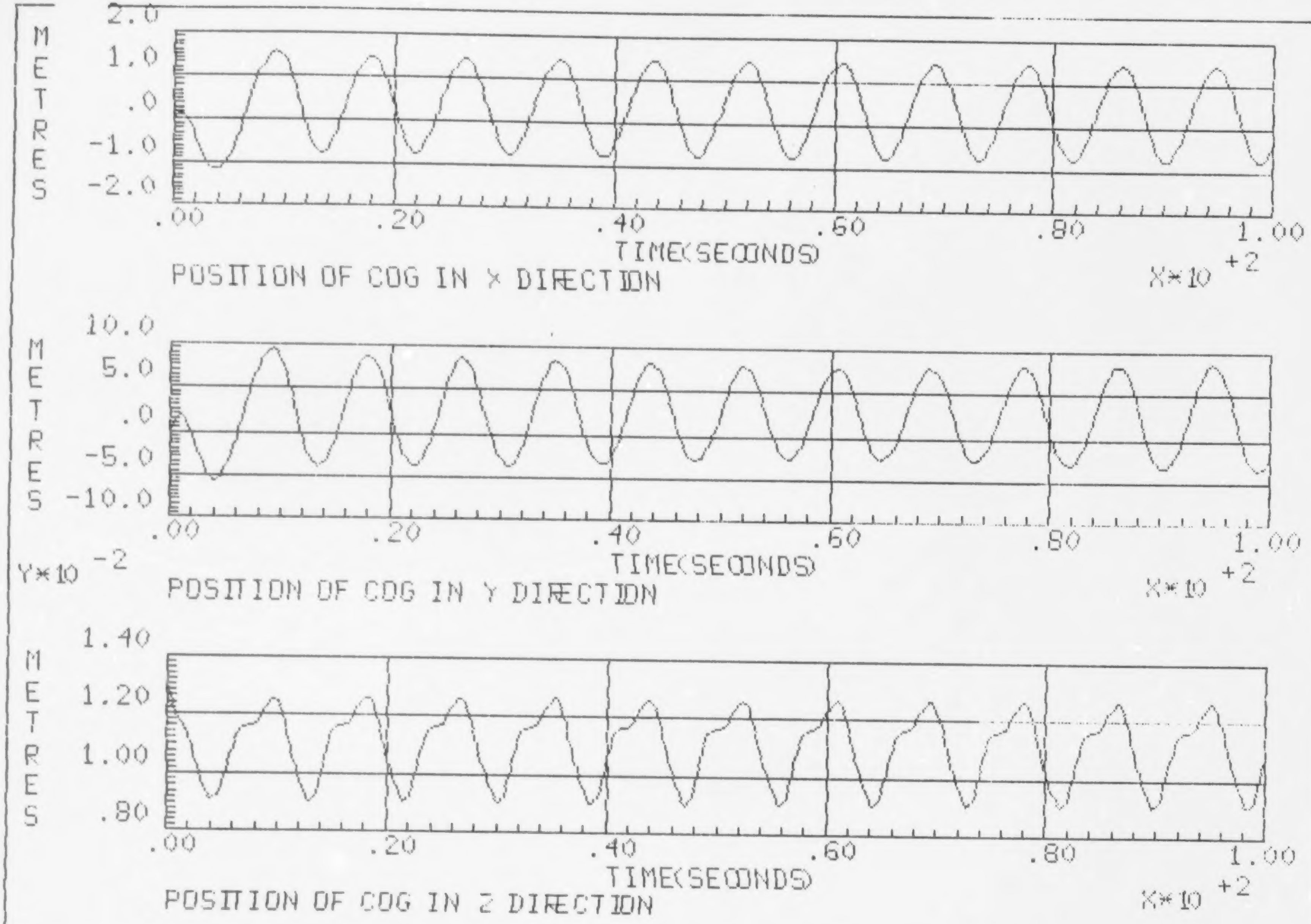
RESULTS AT RESONANCE
ENCOUNTERING FREQUENCY

HARD CHINE VESSEL :

Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
WSM1	5.0	1/12	0.4
WSM2	30.0	1/12	0.4
WSM3	50.0	1/12	0.4
WSM4	5.0	1/7	0.4
WSM5	30.0	1/7	0.4
WSM6	50.0	1/7	0.4
WSM2-10%	30.0	1/12	0.4
WSM3-10%	50.0	1/12	0.4
WSM4-10%	5.0	1/7	0.4
WSM5-10%	30.0	1/7	0.4
WSM6-10%	50.0	1/7	0.4

ROUND BILGE VESSEL :

Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
ESM2-10%	30.0	1/12	10.0
ESM5-10%	30.0	1/7	10.0
ESM6-10%	50.0	1/7	10.0



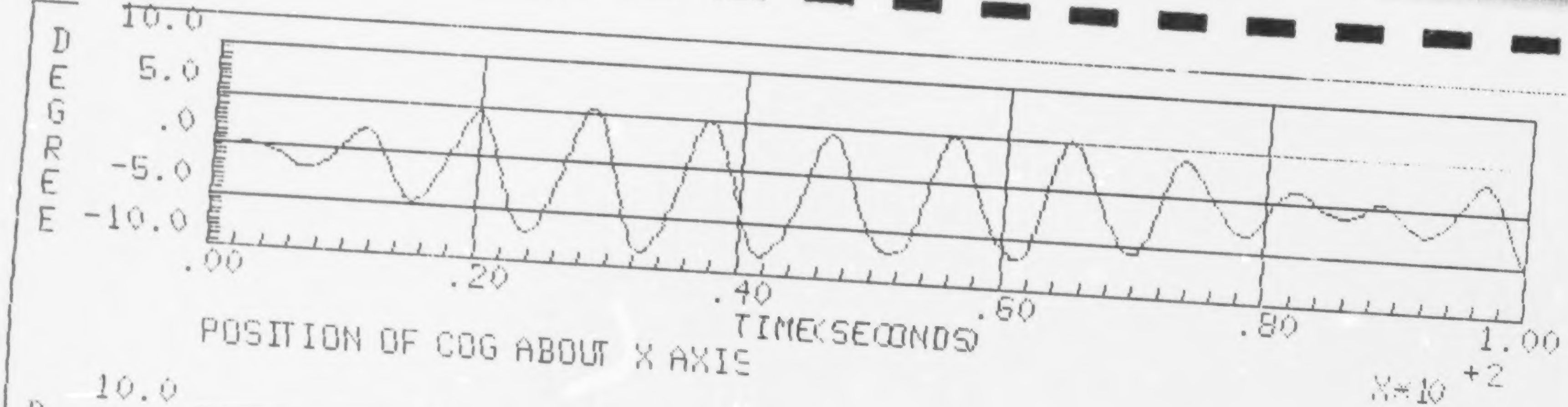
POSITION OF COG IN X DIRECTION

POSITION OF COG IN Y DIRECTION

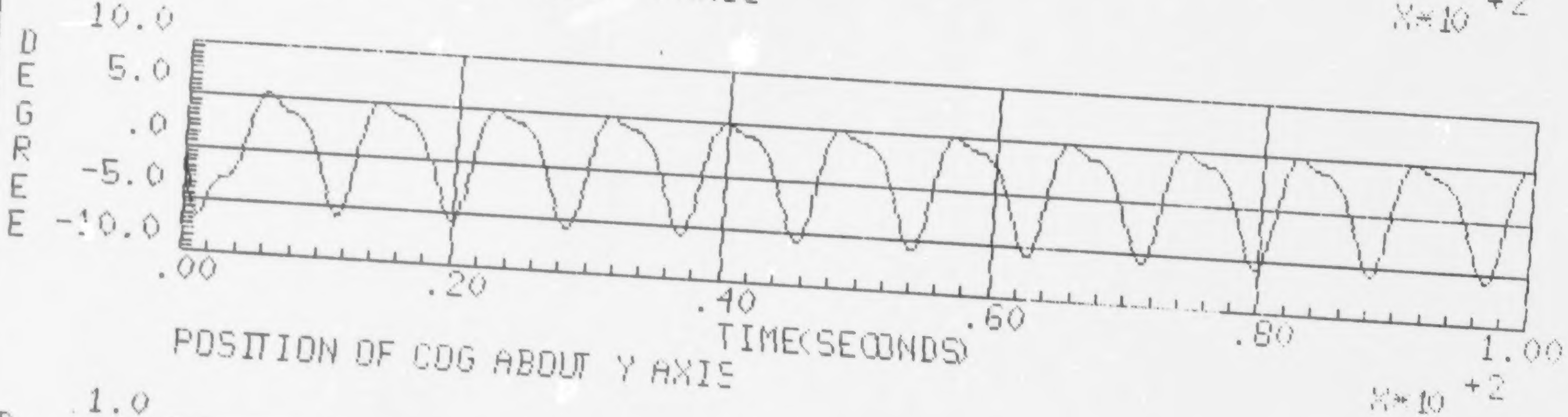
POSITION OF COG IN Z DIRECTION

HARD CHINE VESSEL CASE NO. WSM1

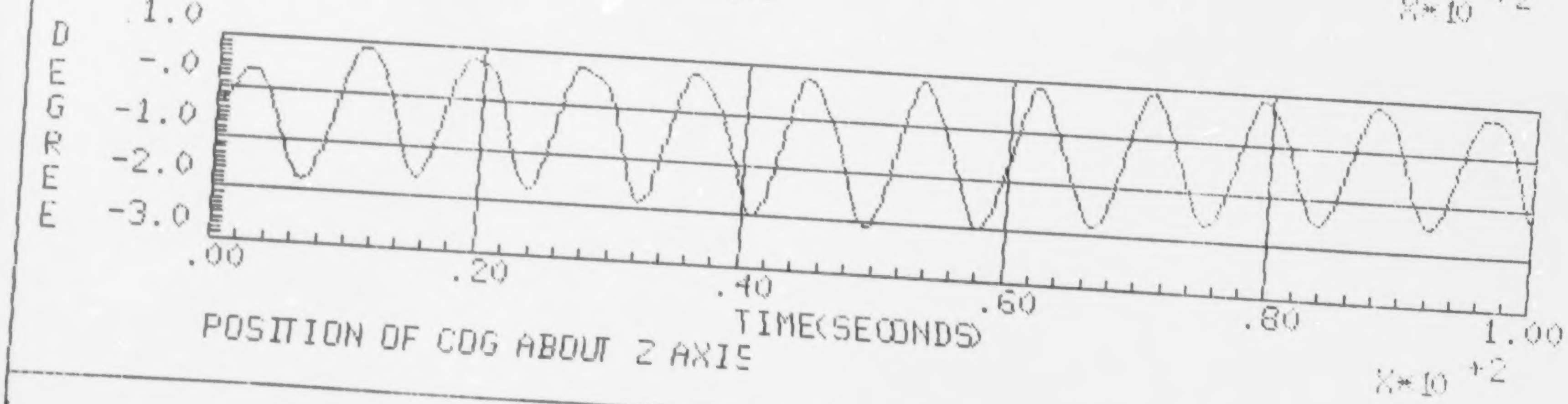
Te=Tr, Dir.=5 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG ABOUT X AXIS



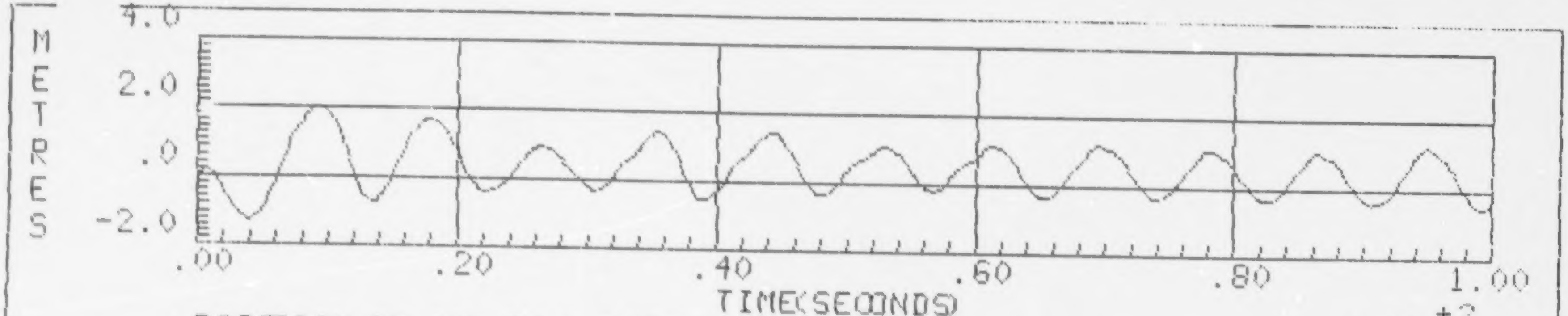
POSITION OF COG ABOUT Y AXIS



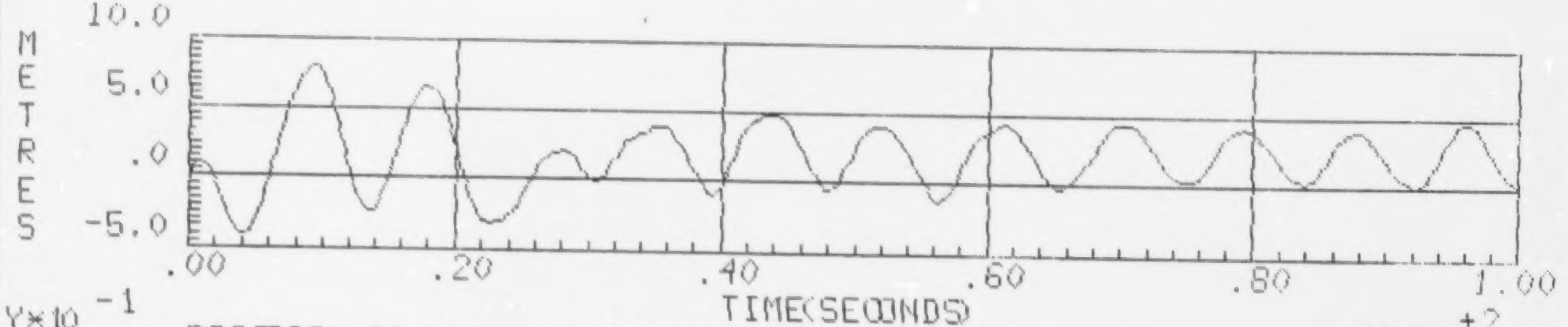
POSITION OF COG ABOUT Z AXIS

HARD CHINE VESSEL. CASE NO. WSM1

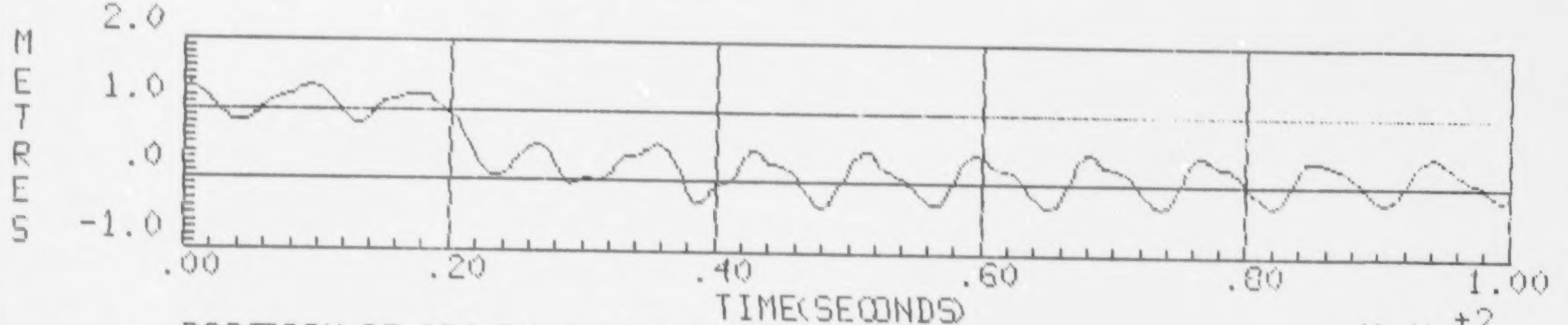
Te=Tr, Dir.=5 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



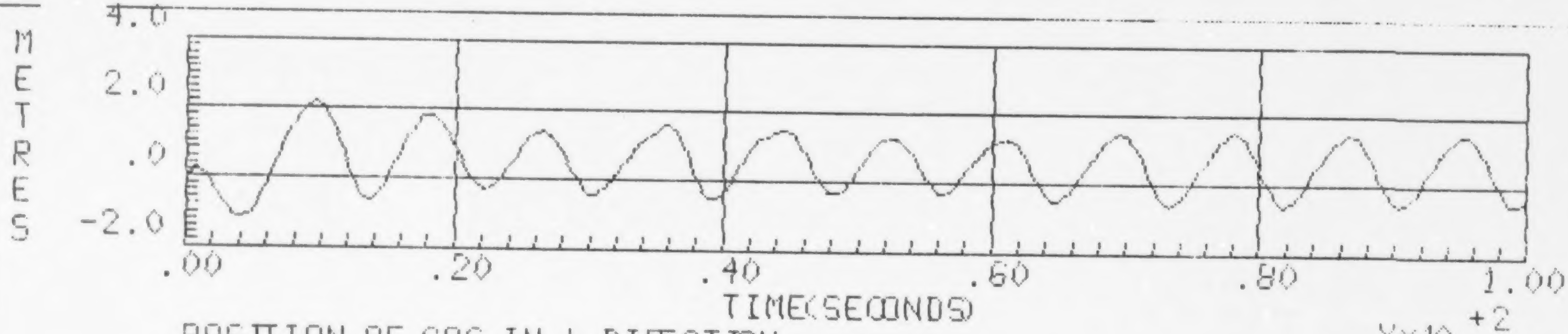
POSITION OF COG IN Y DIRECTION



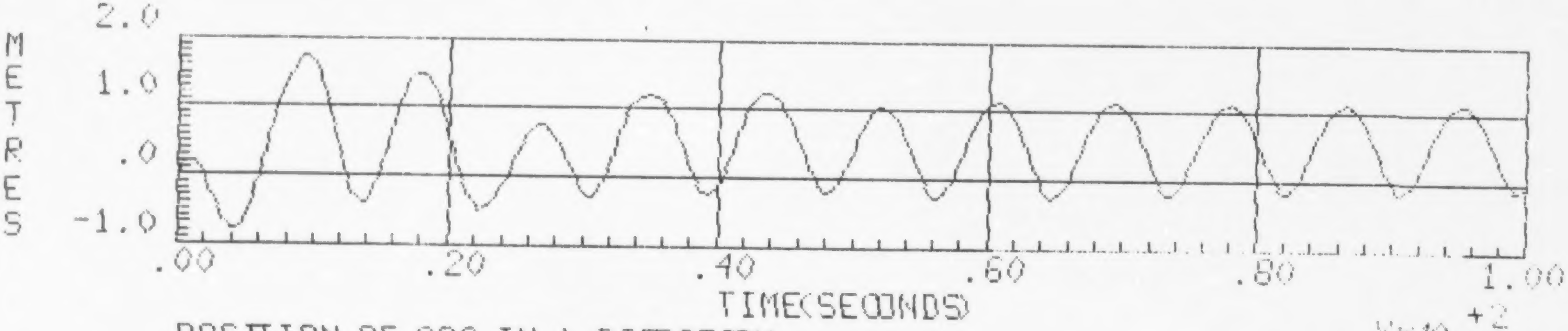
POSITION OF COG IN Z DIRECTION

HARD CHINE VESSEL CASE NO. WSM2

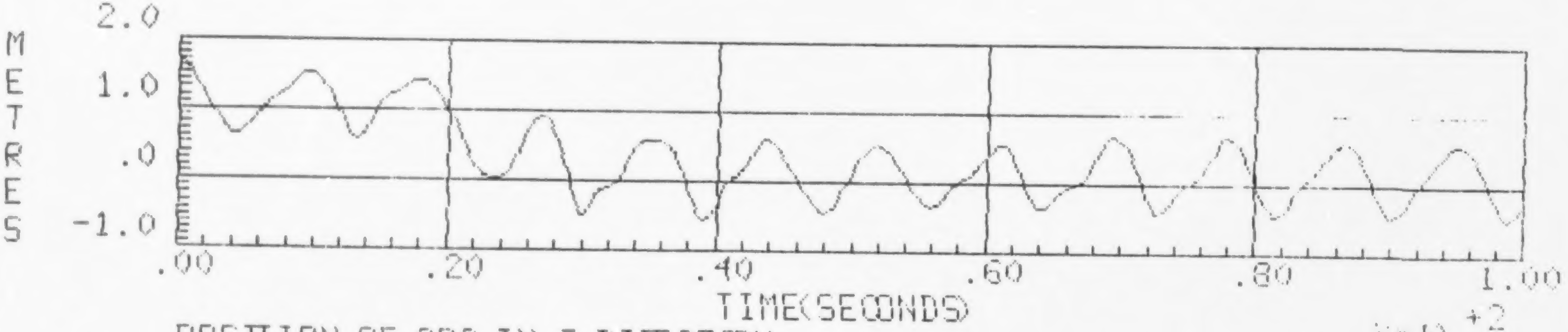
Te=Tr, Dir.=30 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



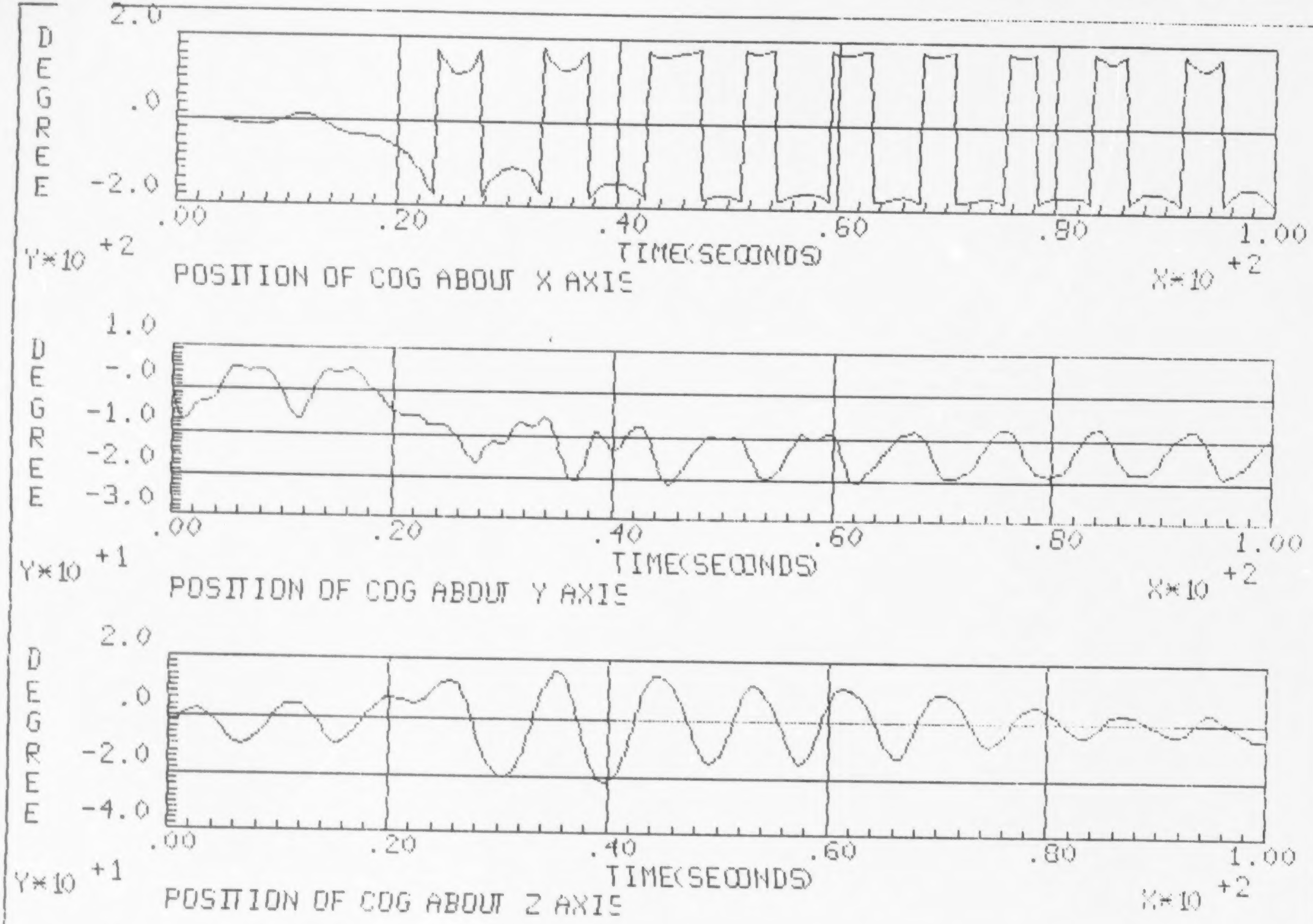
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

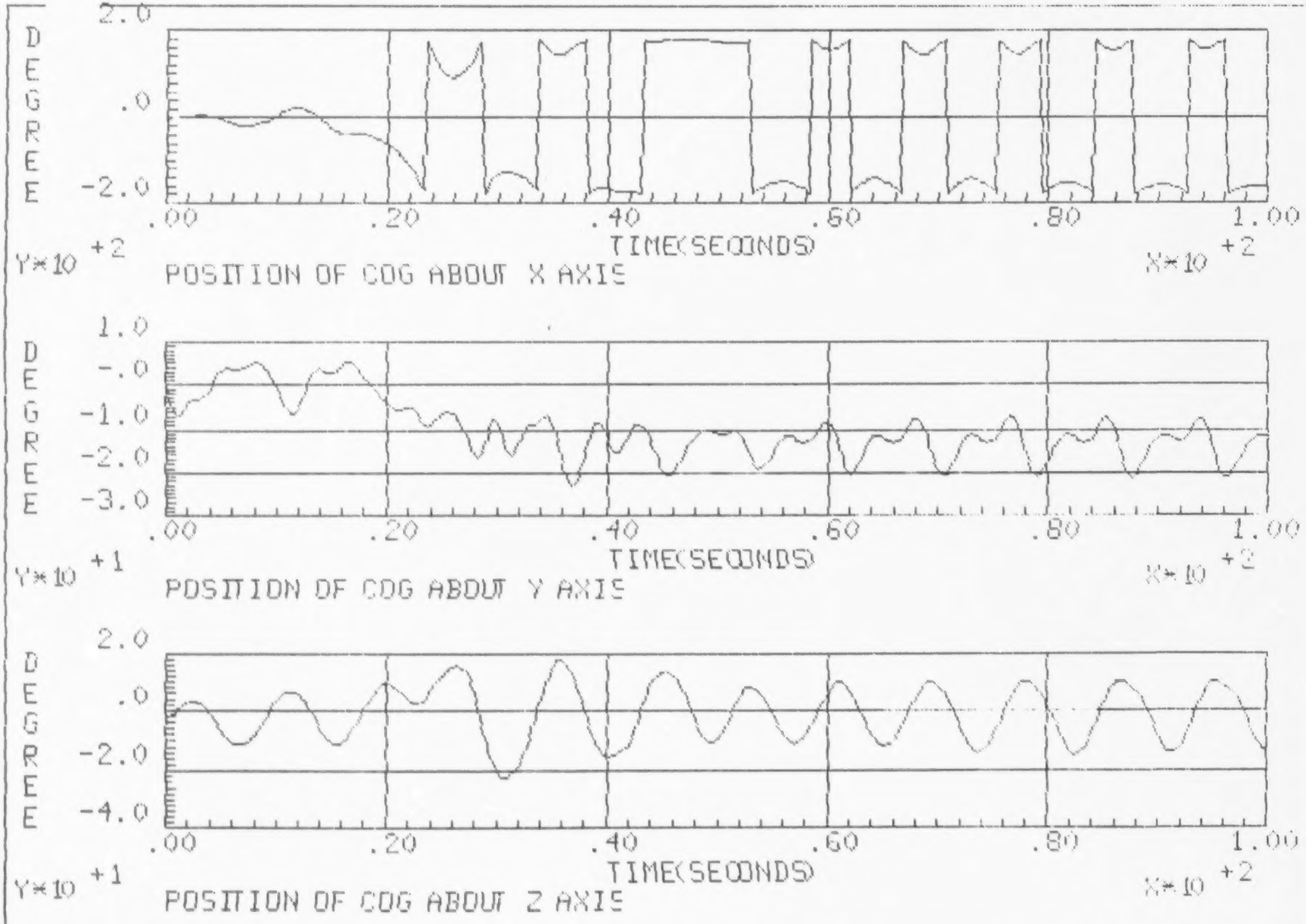
HARD CHINE VESSEL CASE NO. WSM3

Te=Tr, Dir.=50 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



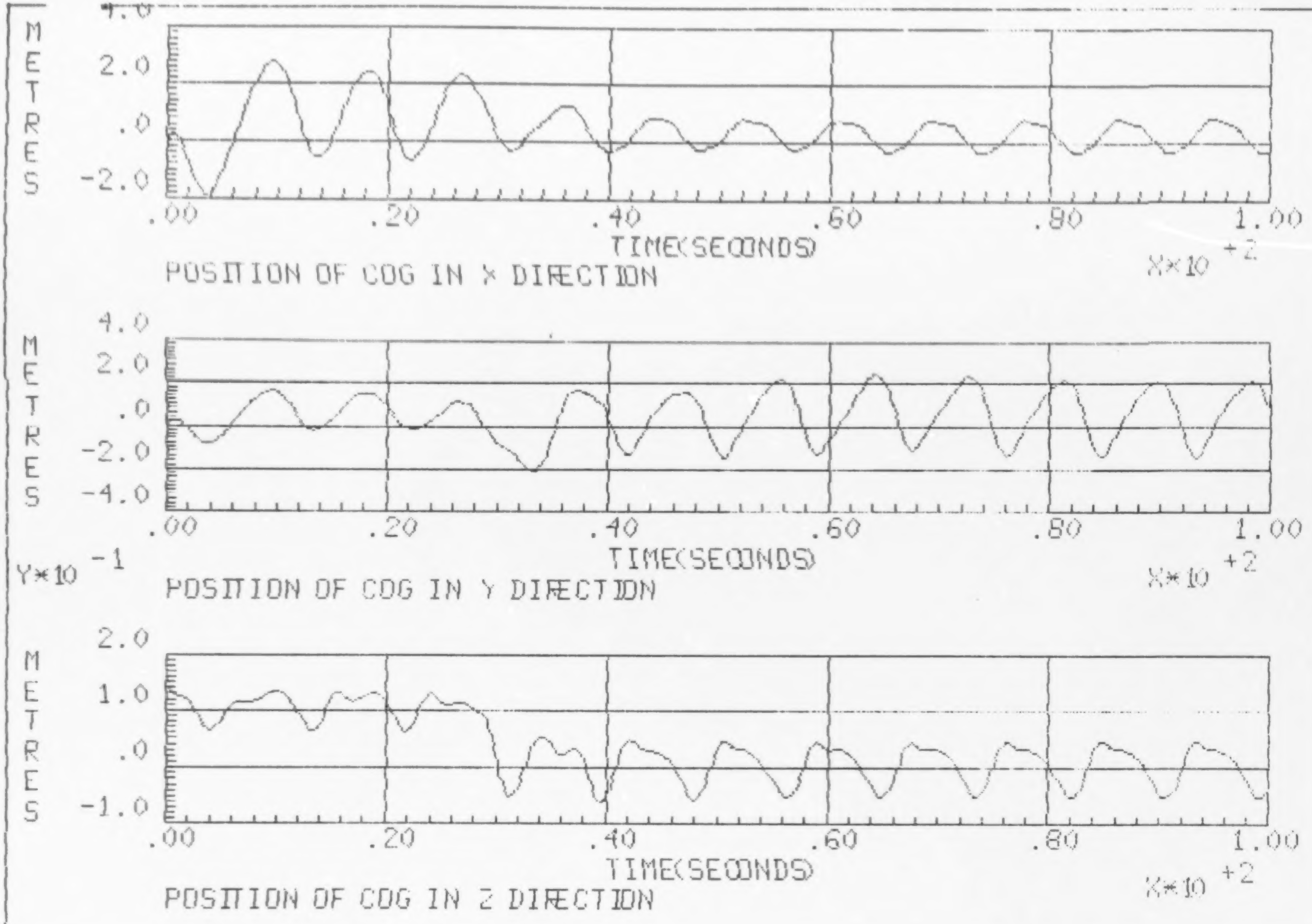
HARD CHINE VESSEL CASE NO. WSM2

Te=Tr, Dir.=30 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



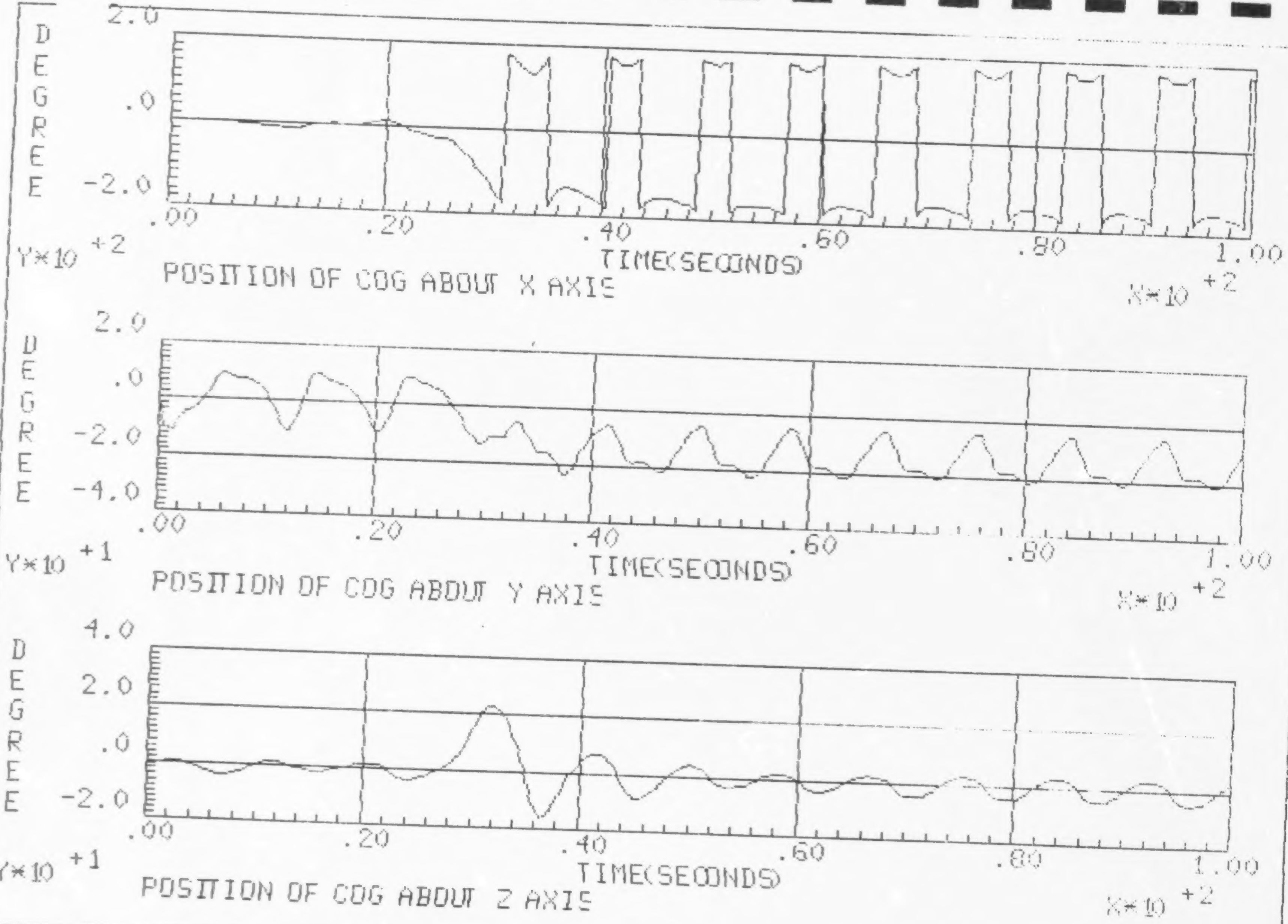
HARD CHINE VESSEL CASE NO. WSM3

Te=Tr, Dir.=50 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



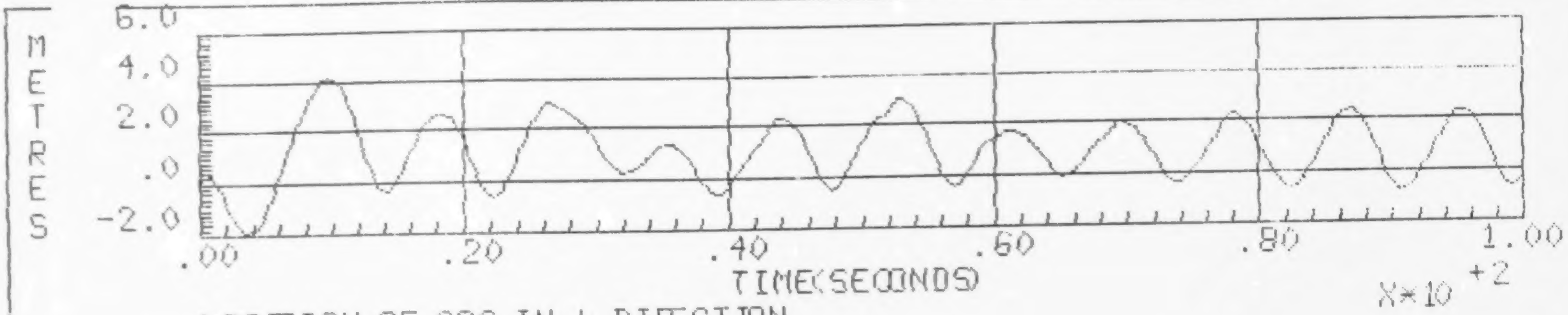
HARD CHINE VESSEL CASE NO. WSM4

Te=Tr, Dir.=5 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2

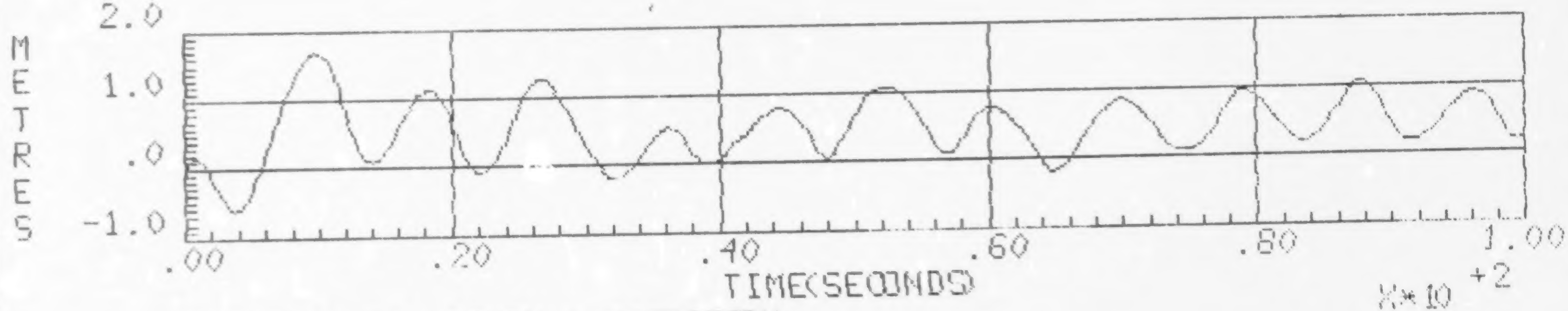


HARD CHINE VESSEL. CASE NO. WSM4

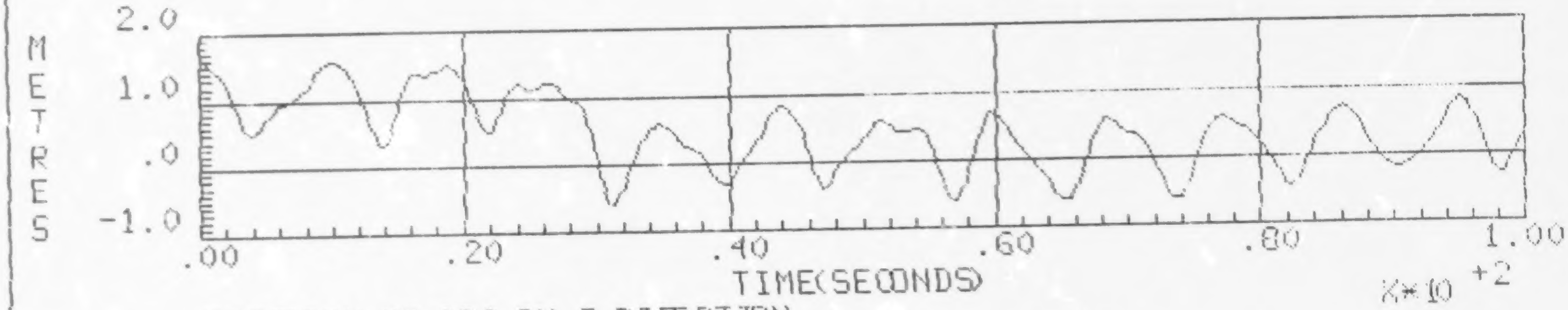
Te=Tr, Dir.=5 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



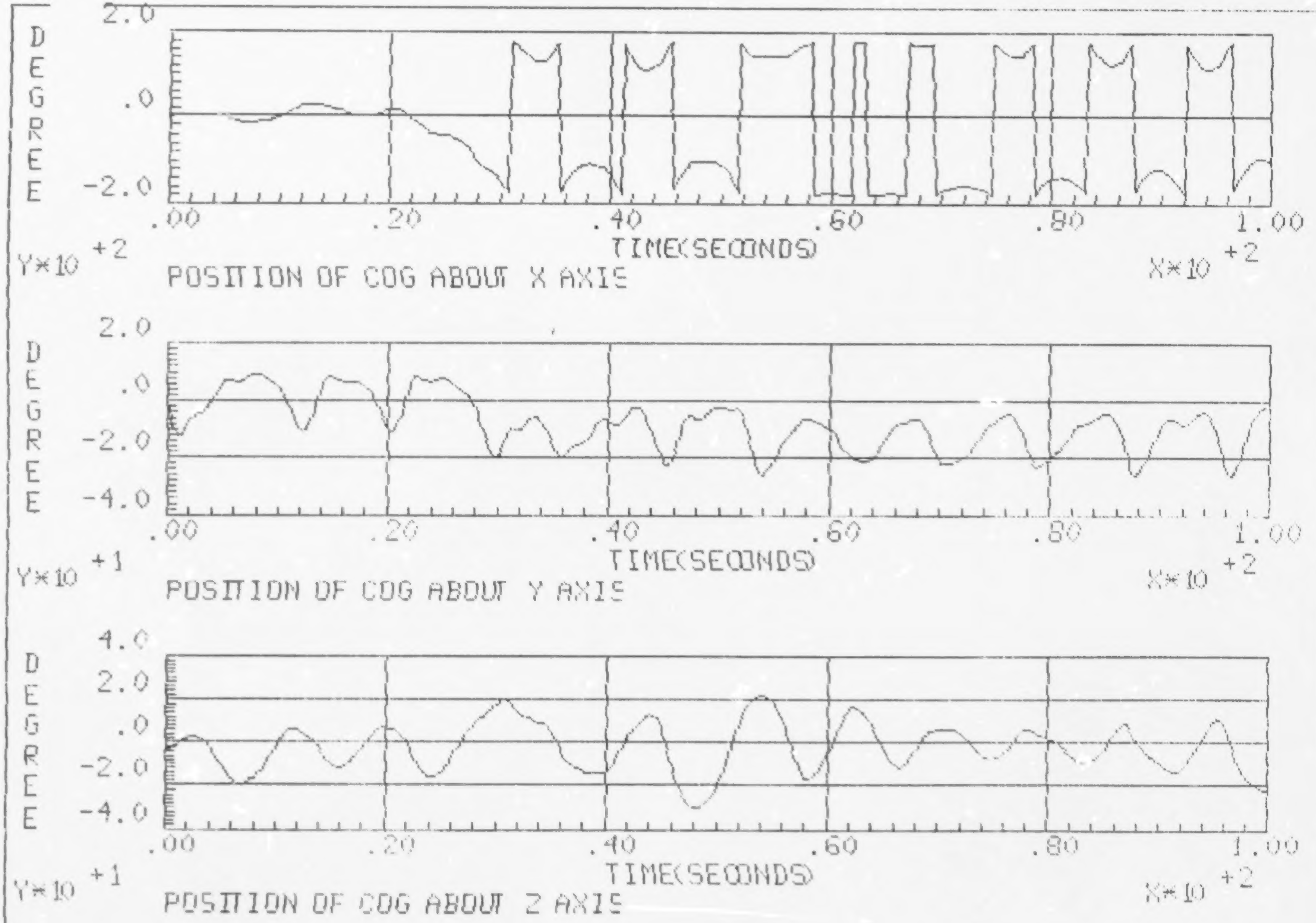
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

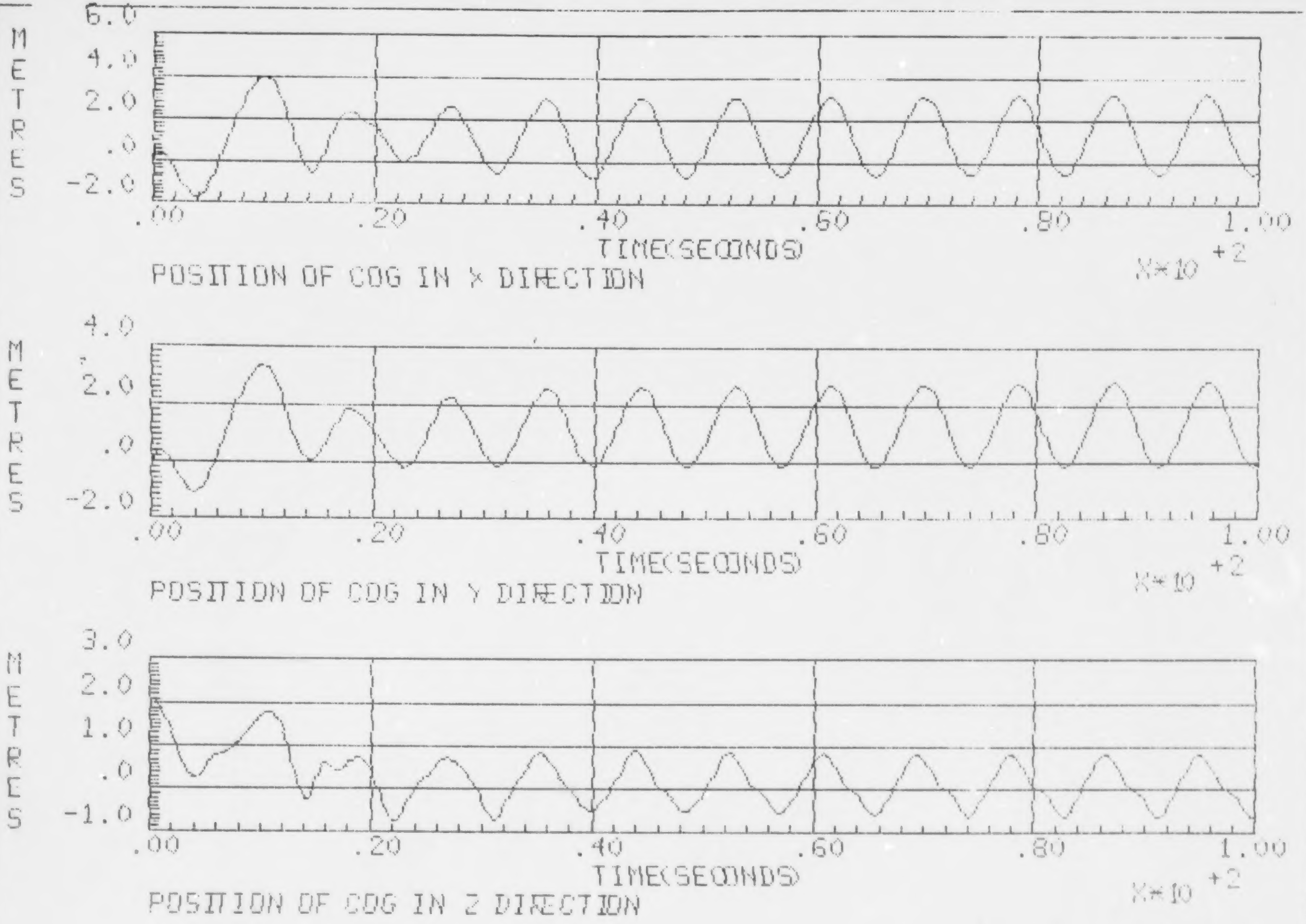
HARD CHINE VESSEL CASE NO. WSM5

Te=Tr, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



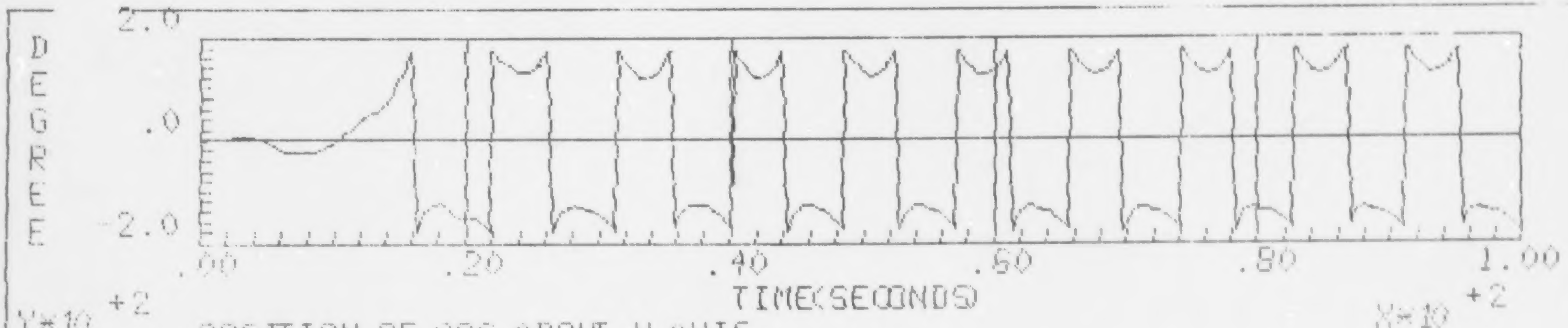
HARD CHINE VESSEL CASE NO. WSM5

Te=Tr, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2

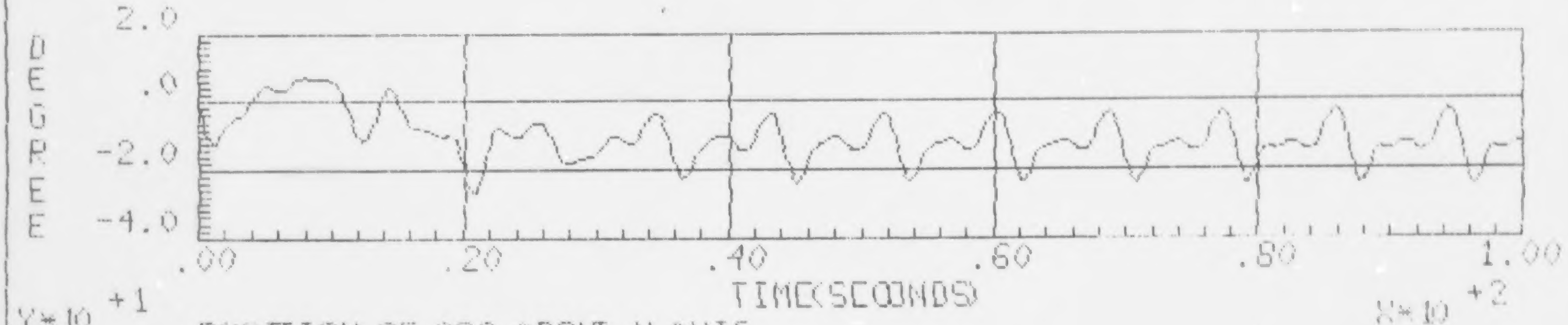


HARD CHINE VESSEL CASE NO. WSM6

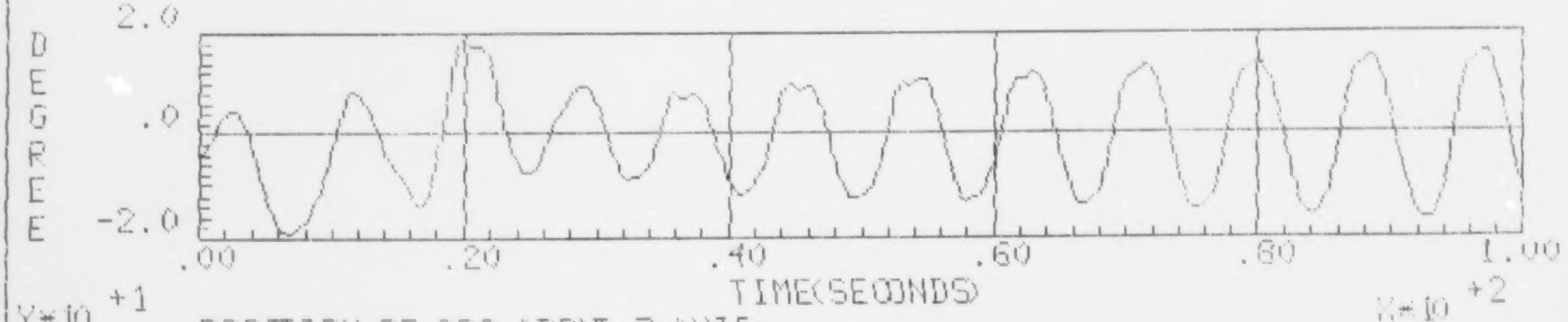
Te=Tr, Dir.=50 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG ABOUT X AXIS



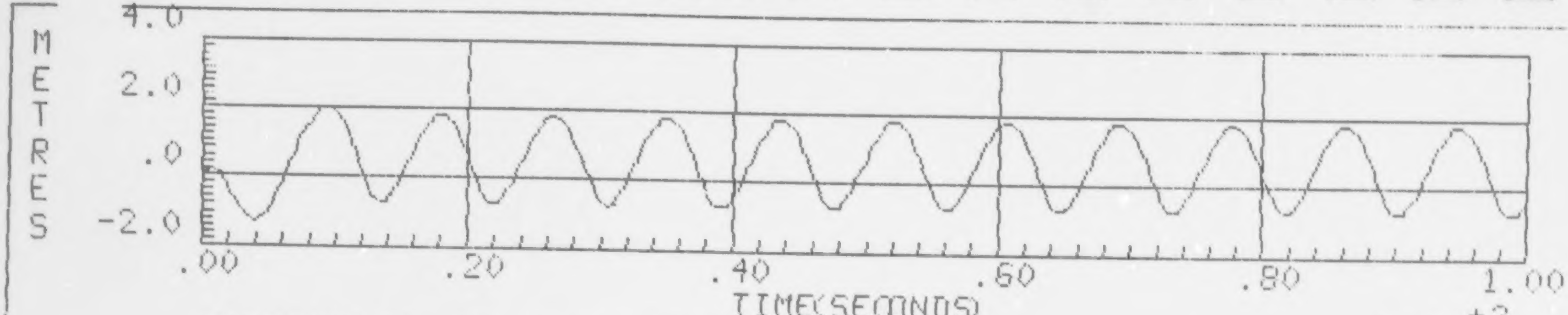
POSITION OF COG ABOUT Y AXIS



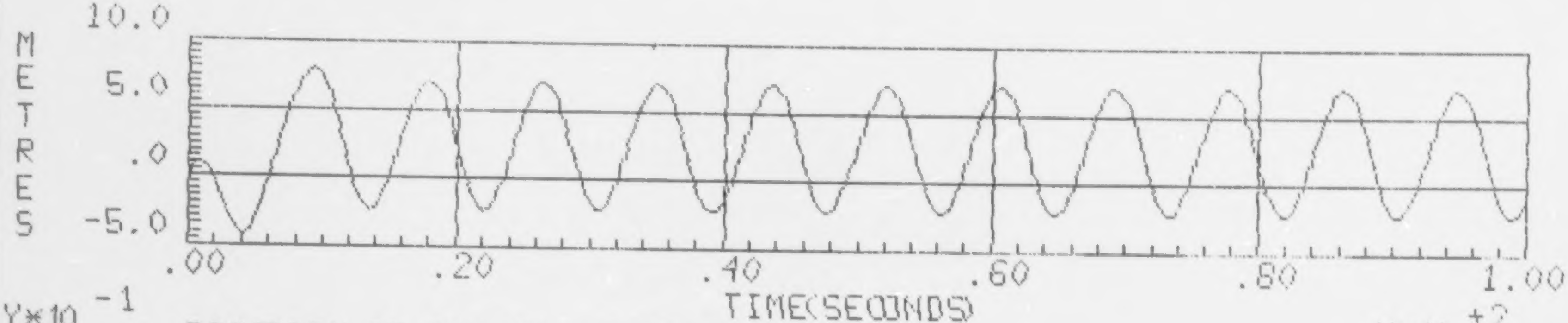
POSITION OF COG ABOUT Z AXIS

HARD CHINE VESSEL CASE NO. WSM6

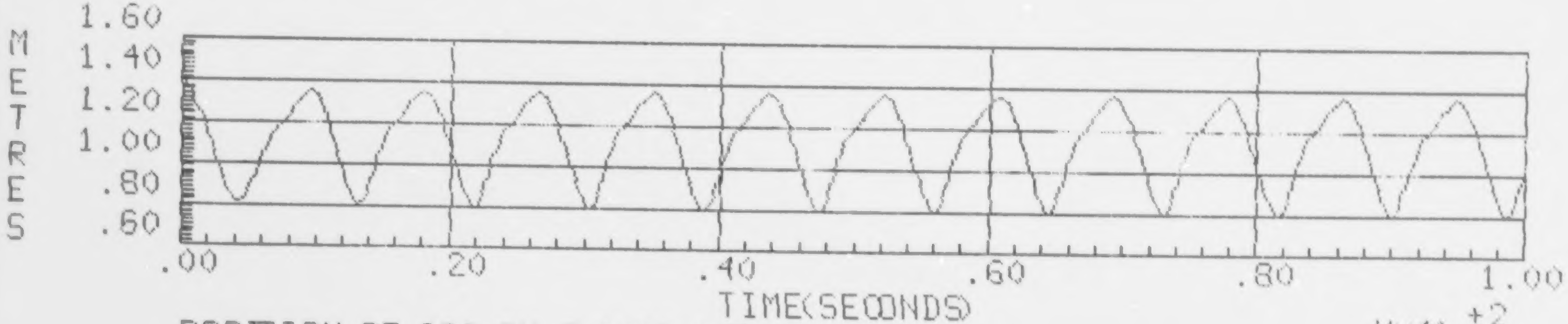
Te=Tr, Dir.=50 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



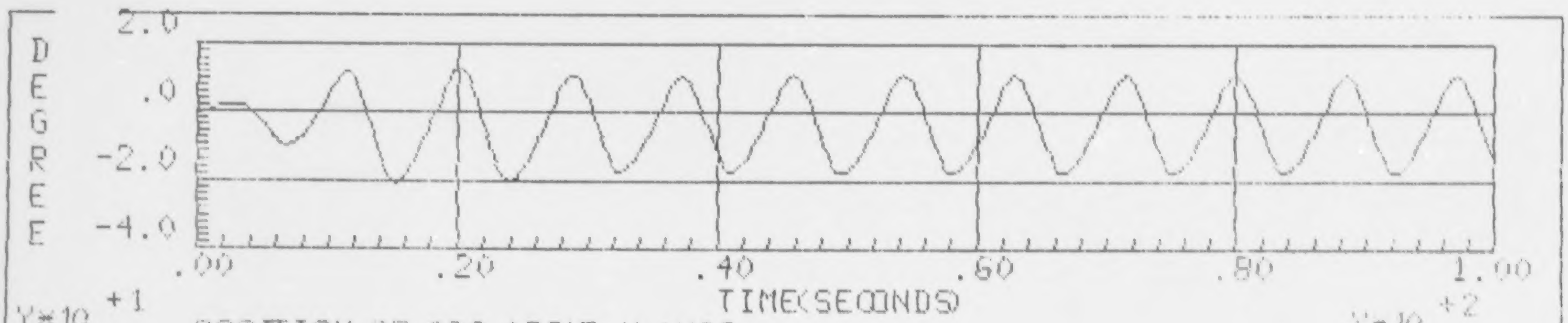
POSITION OF COG IN Y DIRECTION



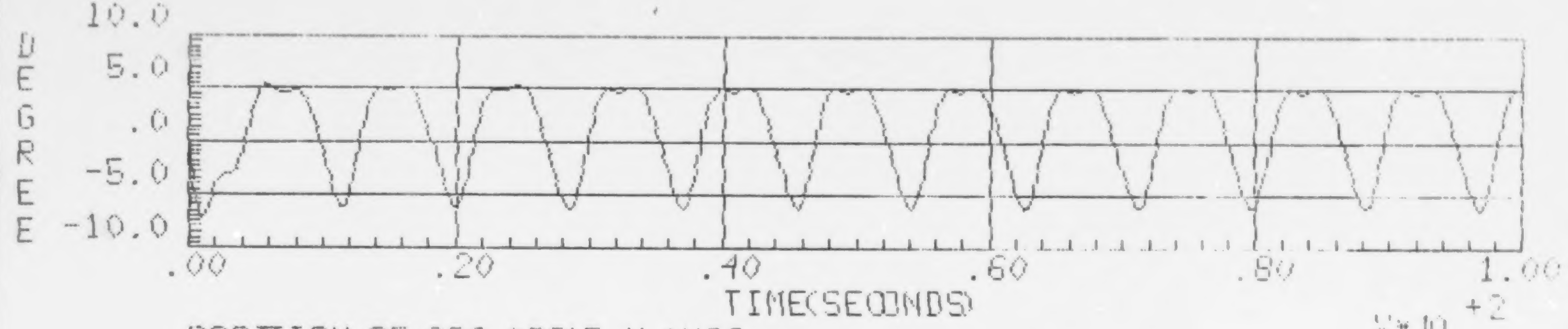
POSITION OF COG IN Z DIRECTION

HARD CHINE VESSEL. CASE NO. WSM2-10%

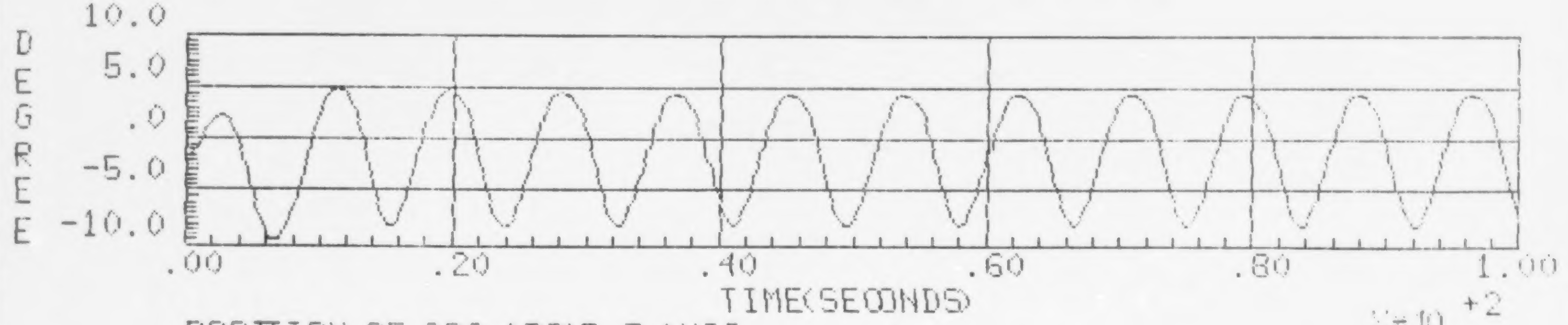
Te=Tr, Dir.=30 °, H=1.91 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



POSITION OF COG ABOUT X AXIS



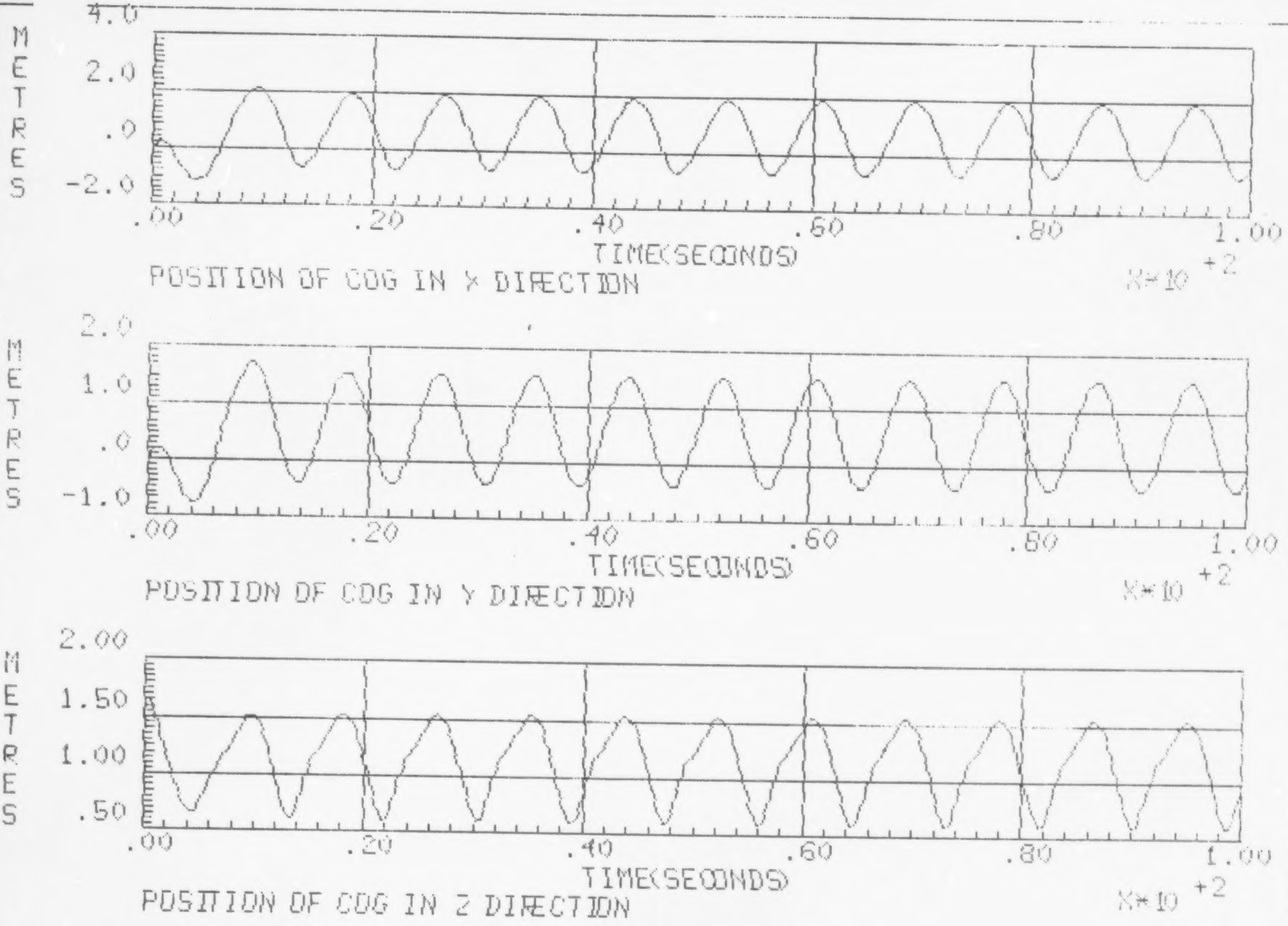
POSITION OF COG ABOUT Y AXIS



POSITION OF COG ABOUT Z AXIS

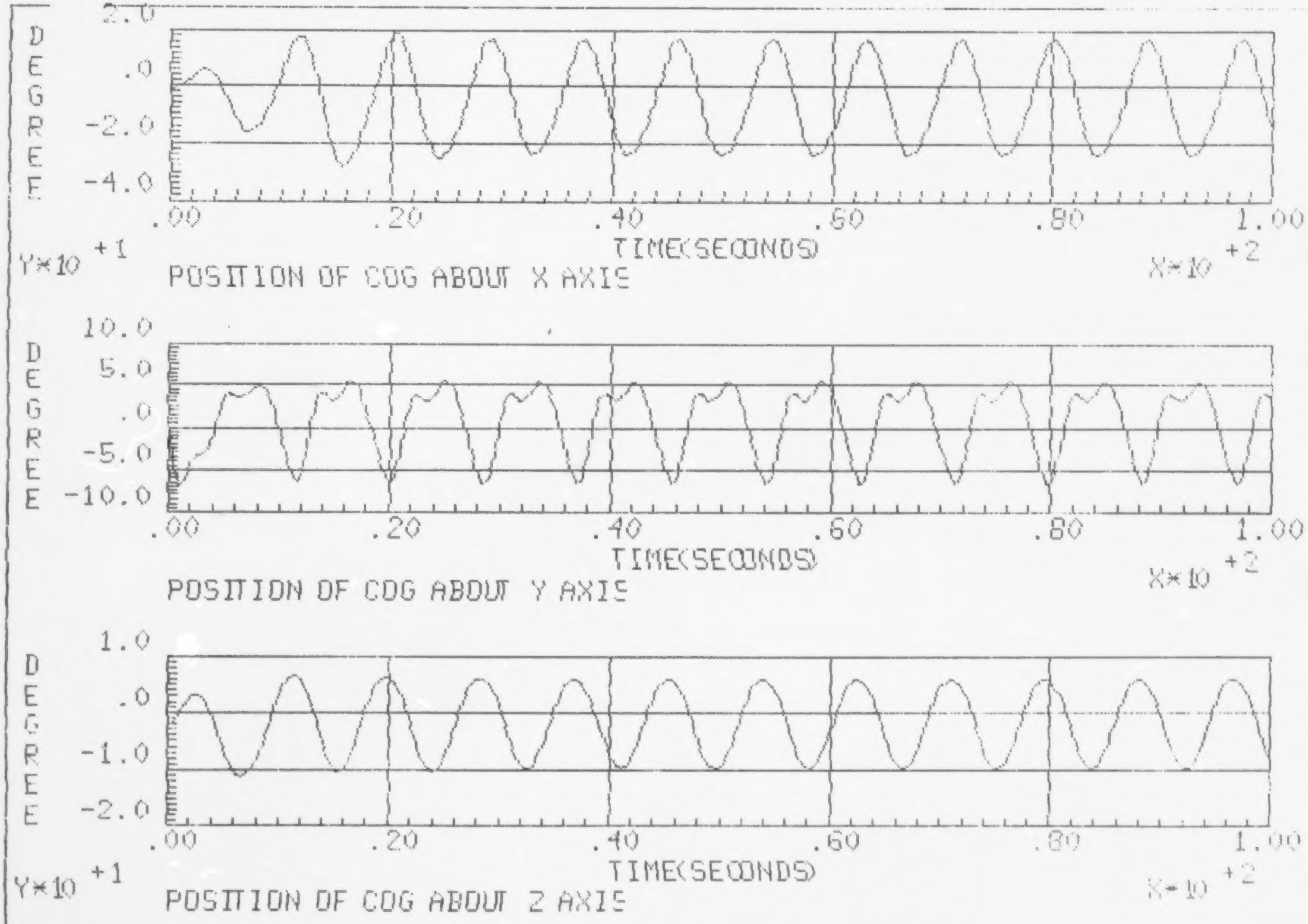
HARD CHINE VESSEL CASE NO. WSM2-10%

Te=Tr, Dir.=30 °, H=1.91 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



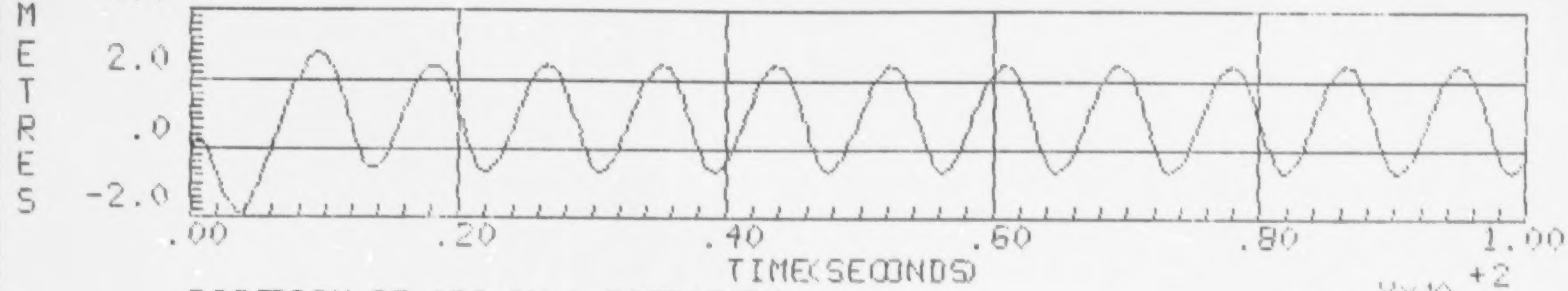
HARD CHINE VESSEL CASE NO. WSM3-10%

Te=Tr, Dir.=50 °, H=1.91 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2

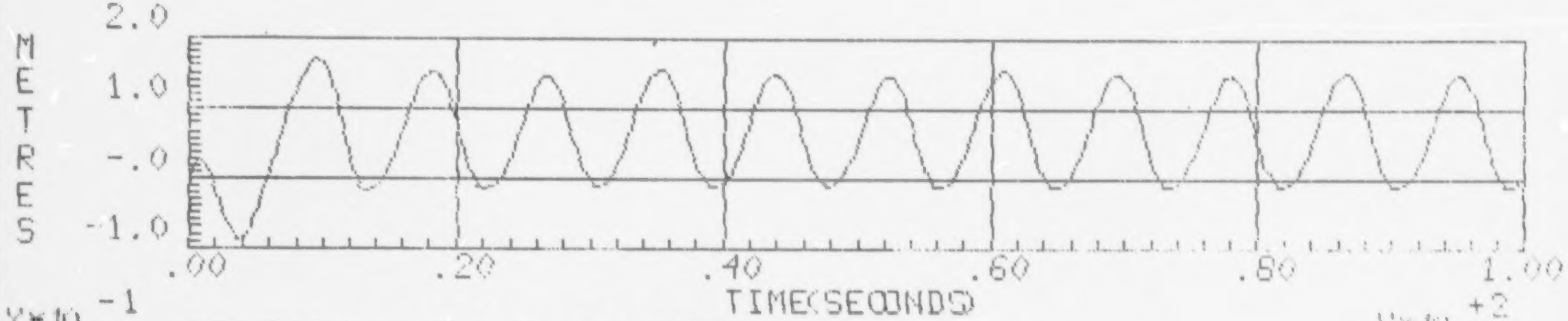


HARD CHINE VESSEL. CASE NO. WSM3-10%

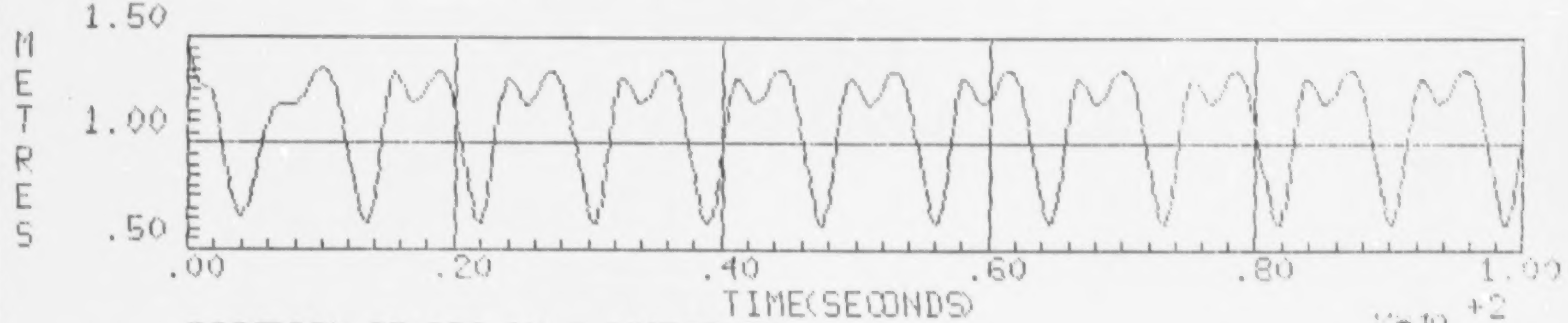
Te=Tr, Dir.=50 °, H=1.91 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



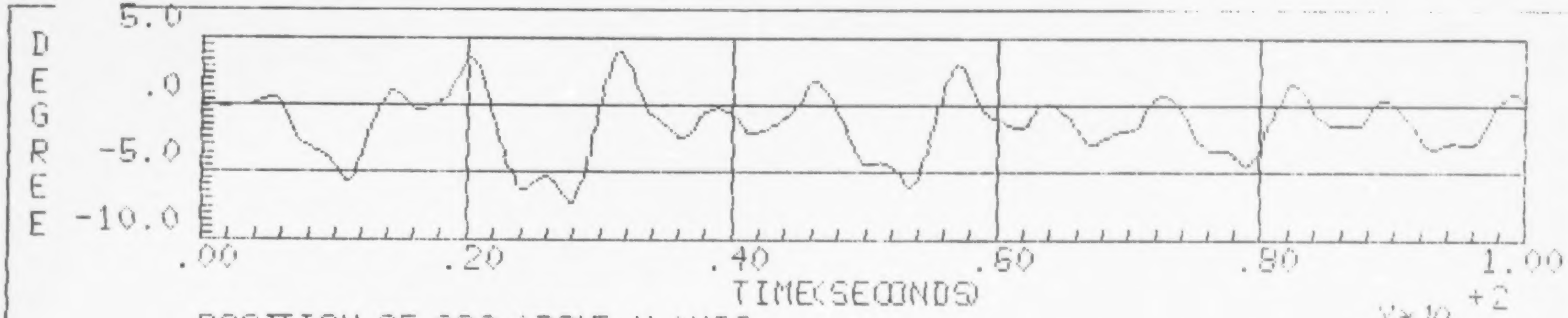
POSITION OF COG IN Y DIRECTION



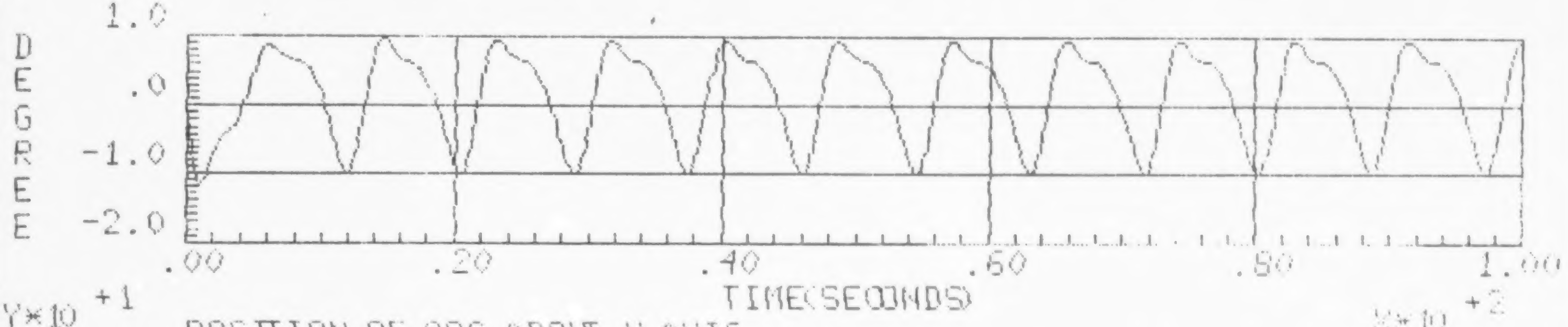
POSITION OF COG IN Z DIRECTION

HARD CHINE VESSEL CASE NO. WSM4-10%

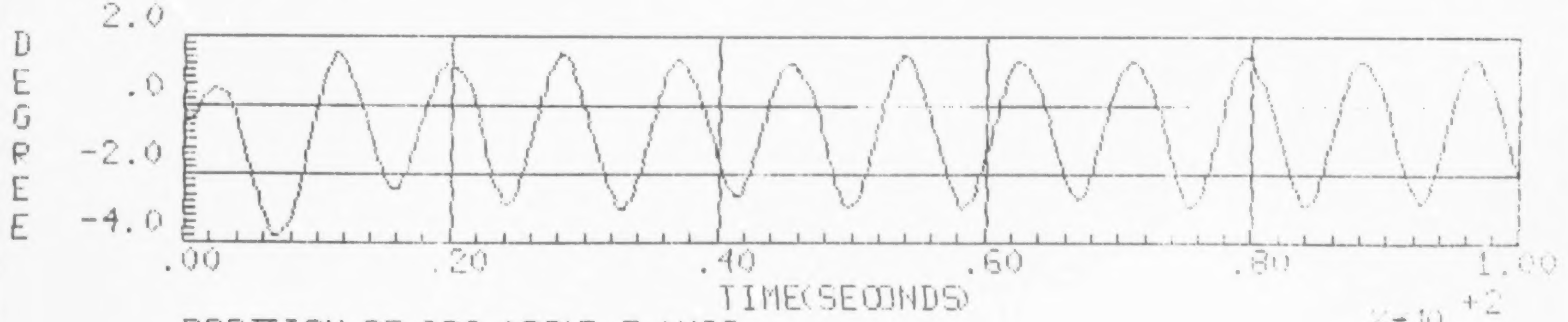
Te=Tr, Dir.=5 °, H=3.24 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



POSITION OF COG ABOUT X AXIS



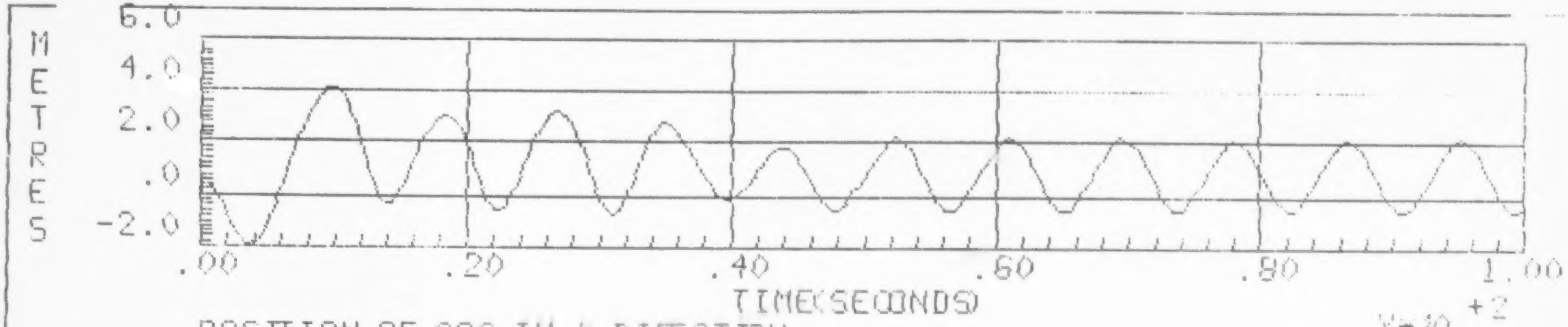
POSITION OF COG ABOUT Y AXIS



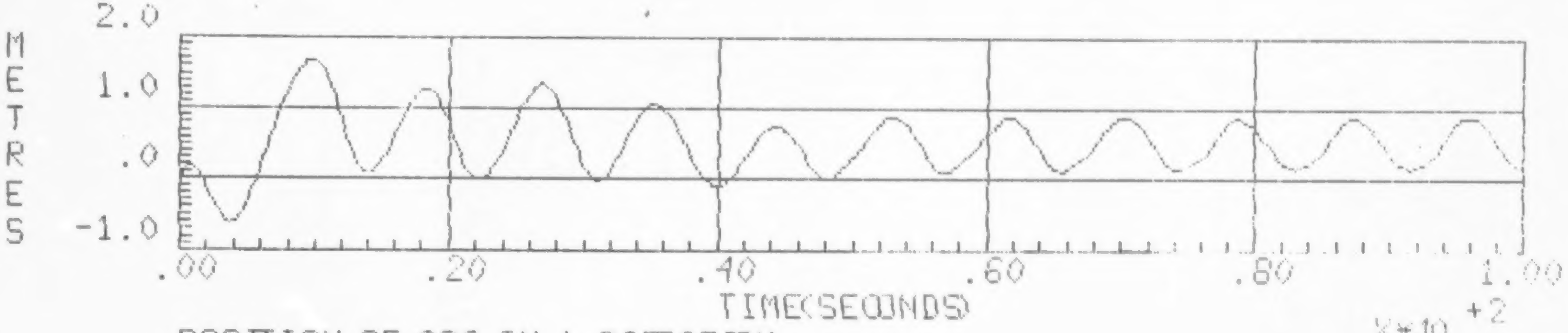
POSITION OF COG ABOUT Z AXIS

HARD CHINE VESSEL CASE NO. WSM4-10%

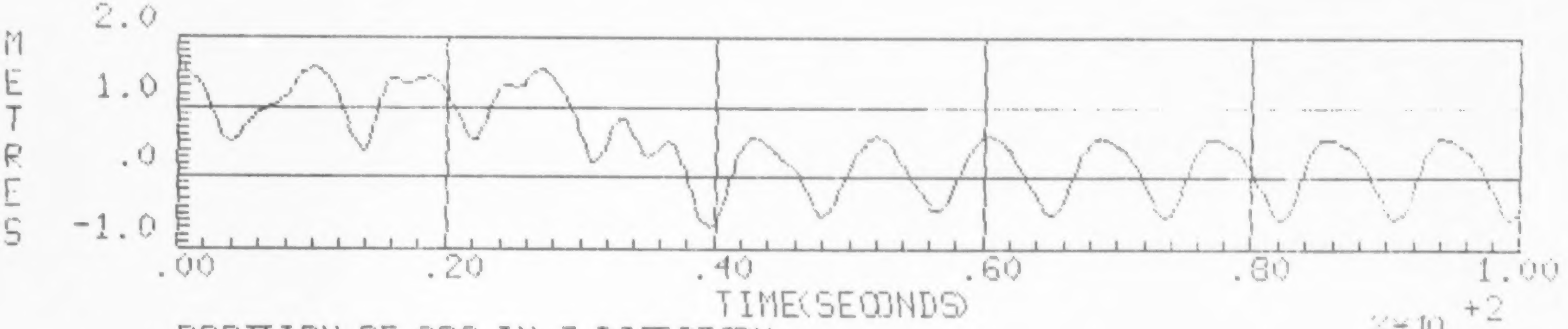
Te=Tr, Dir.=5 °, H=3.24 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



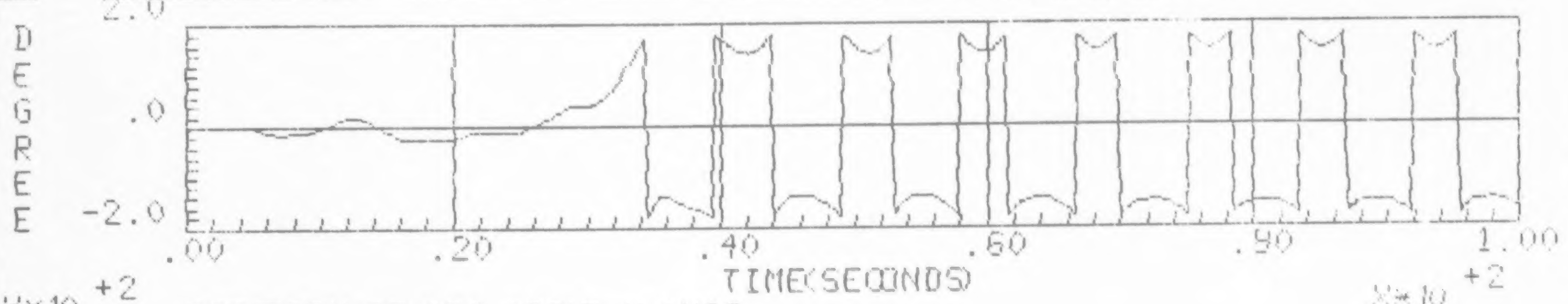
POSITION OF COG IN Y DIRECTION



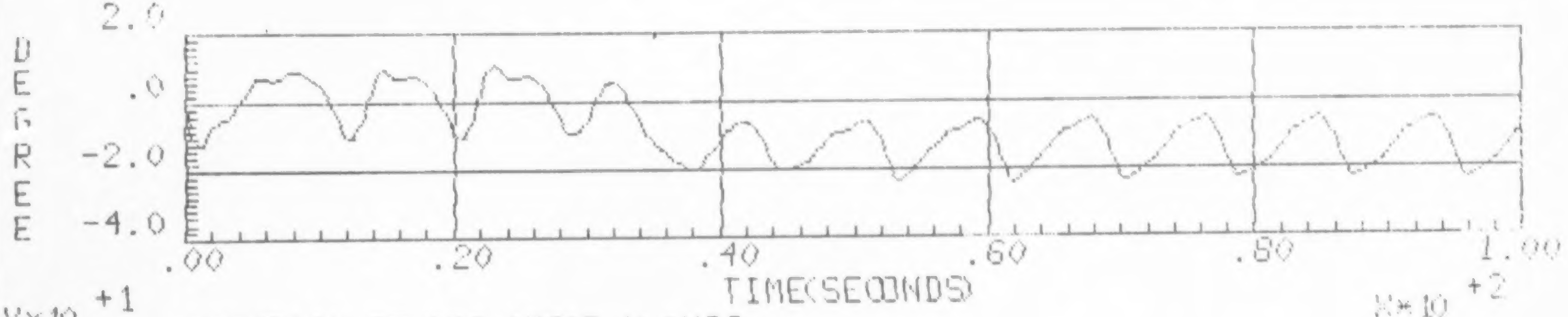
POSITION OF COG IN Z DIRECTION

HARD CHINE VESSEL CASE NO. WSM5-10%

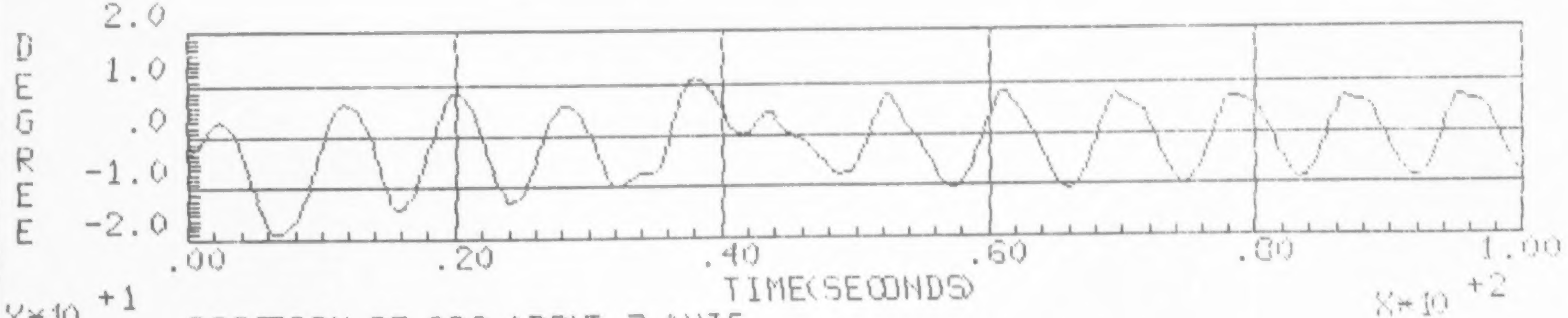
Te=Tr, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



POSITION OF COG ABOUT X AXIS



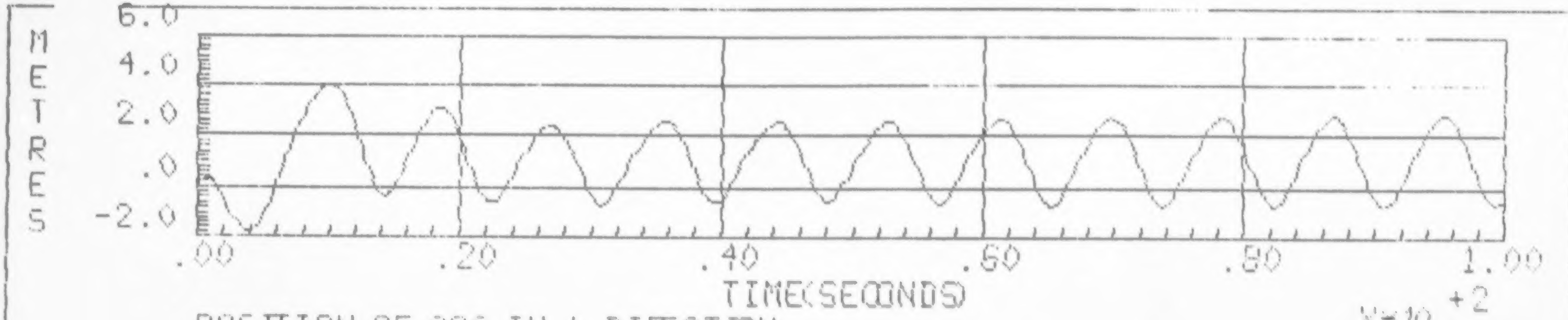
POSITION OF COG ABOUT Y AXIS



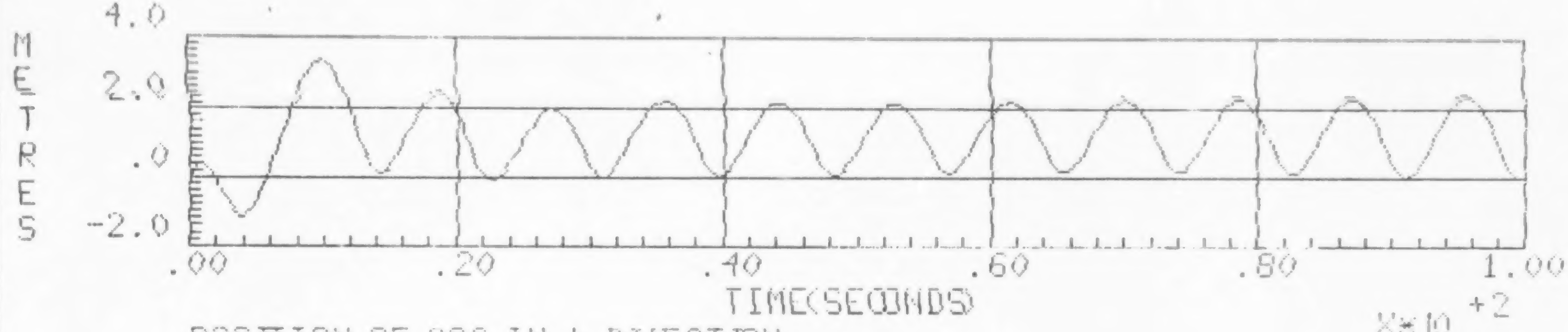
POSITION OF COG ABOUT Z AXIS

HARD CHINE VESSEL CASE NO. WSM5-10%

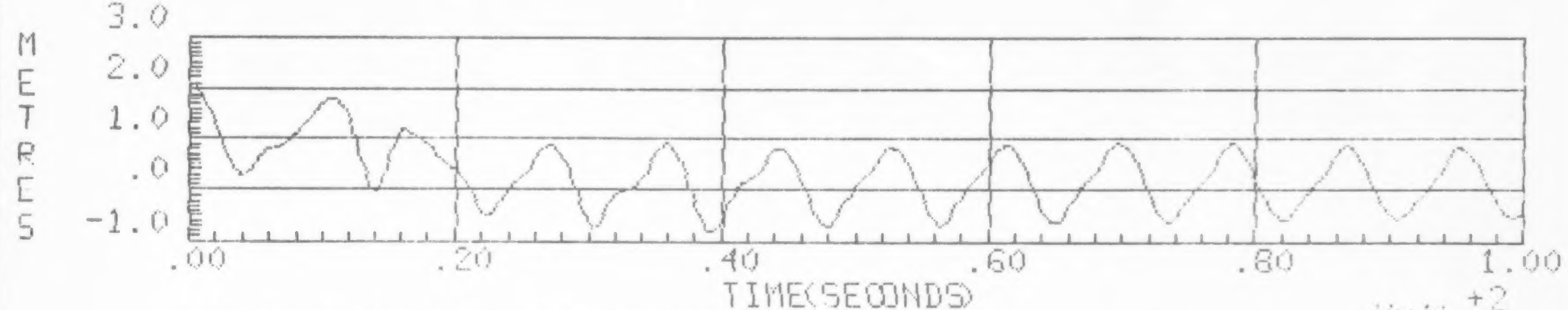
Te=Tr, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



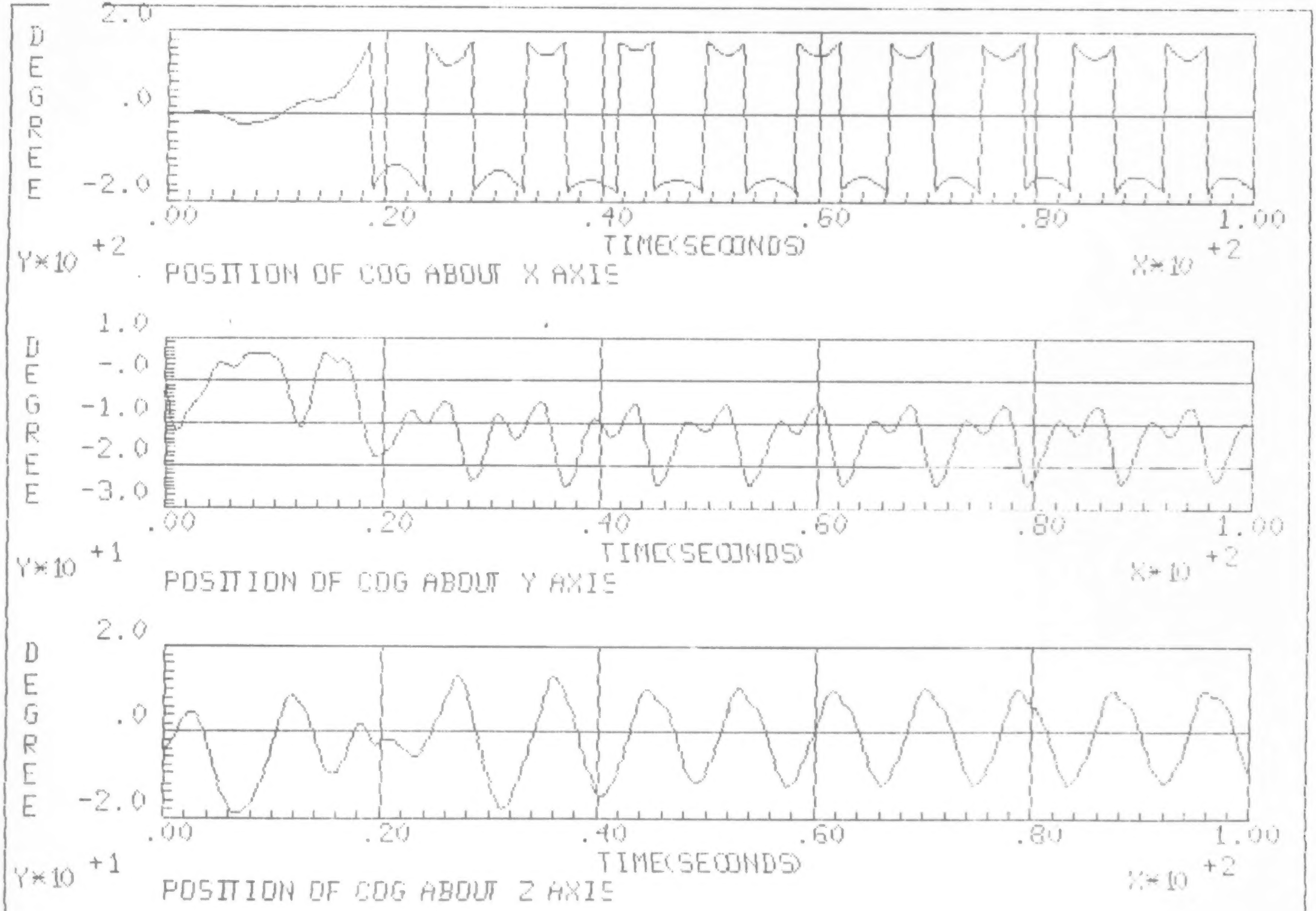
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

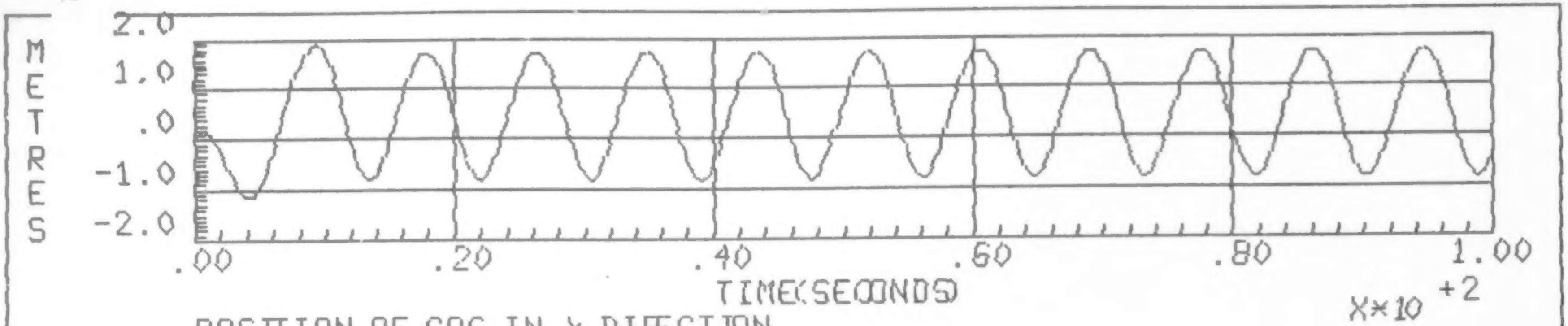
HARD CHINE VESSEL CASE NO. WSM6-10%

Te=Tr, Dir.=50 °, H=3.24 m, Diff.=0, D_{44}=10.0% CRIT., Servo 2

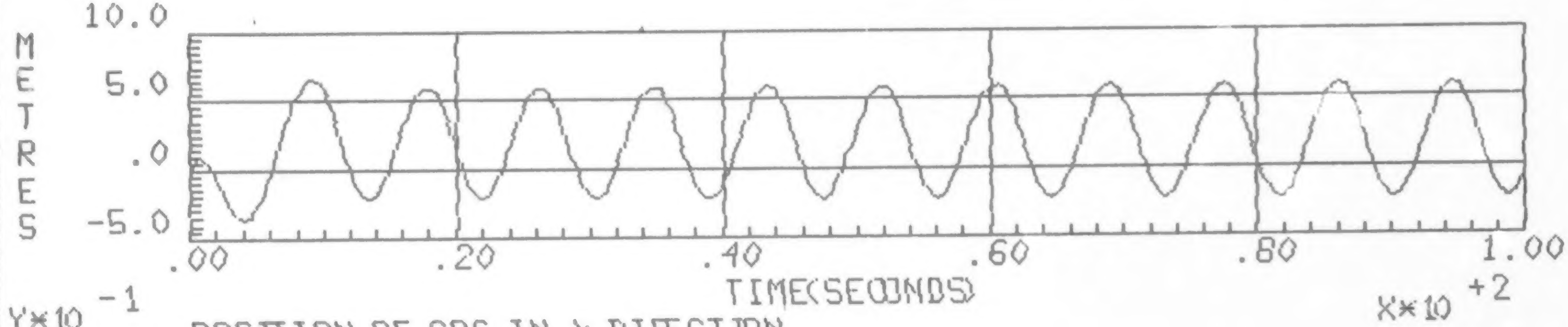


HARD CHINE VESSEL CASE NO. WSM6-10%

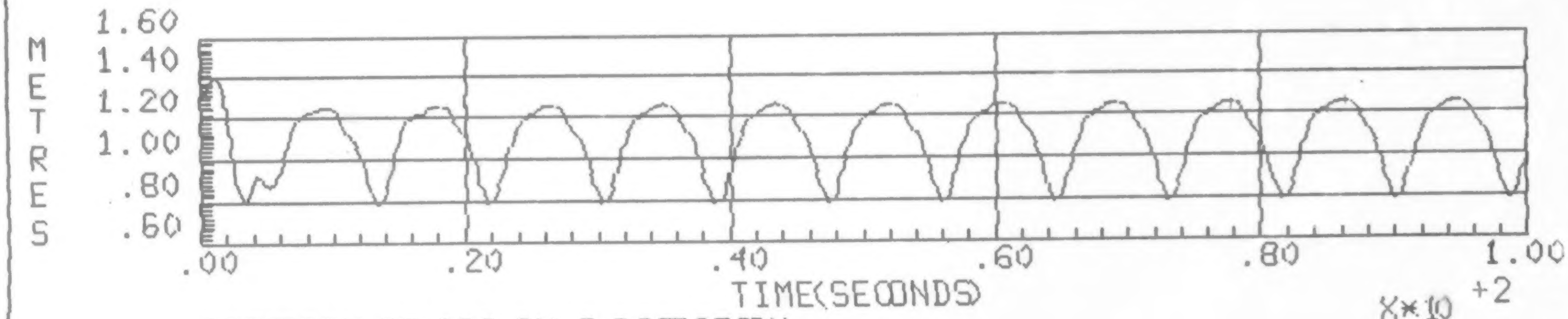
Te=Tr, Dir.=50 °, H=3.24 m, Diff.=0, D_{44}=10.0% CRIT., Servo 2



POSITION OF COG IN X DIRECTION



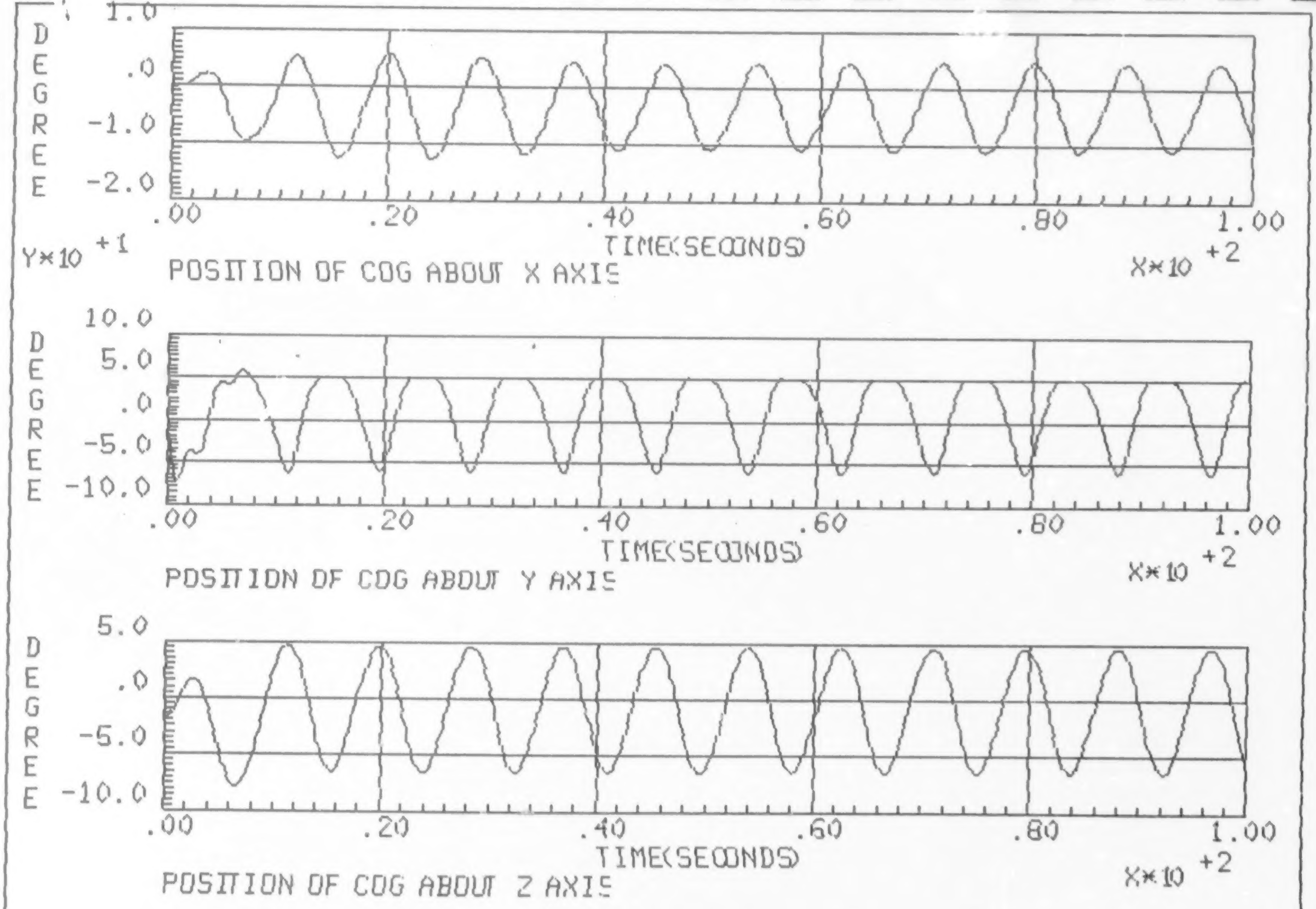
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

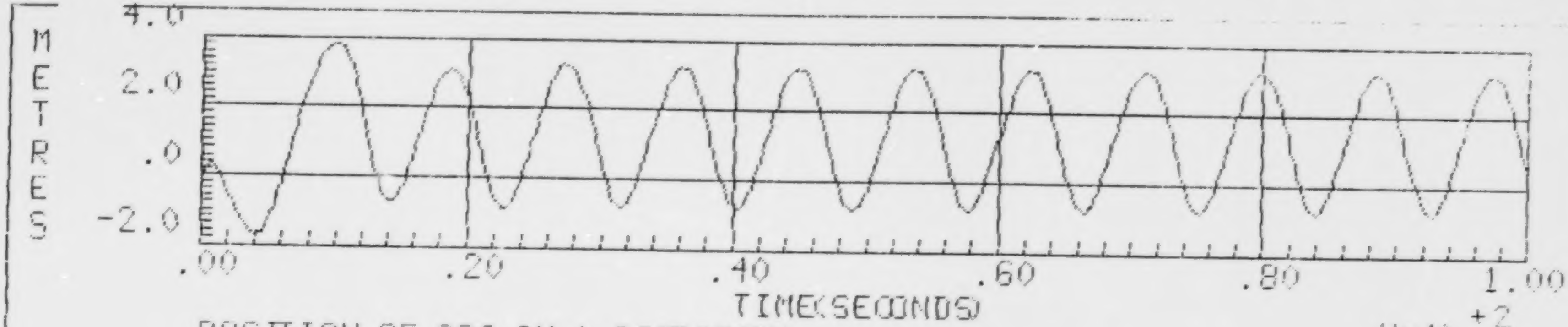
ROUND BILGE VESSEL CASE NO. ESM2-10%

Te=Tr, Dir.=30 °, H=1.91 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2

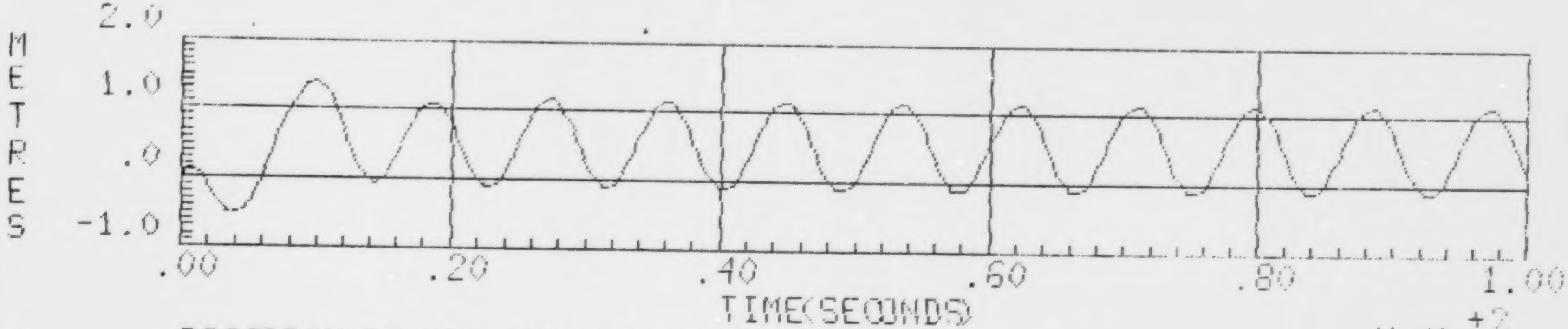


ROUND BILGE VESSEL CASE NO. ESM2-10%

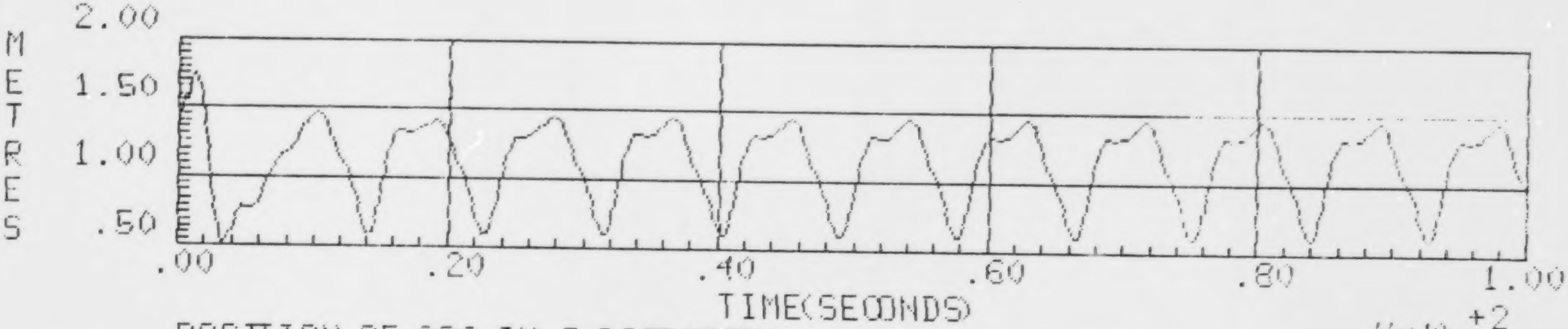
Te=Tr, Dir.=30 °, H=1.91 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



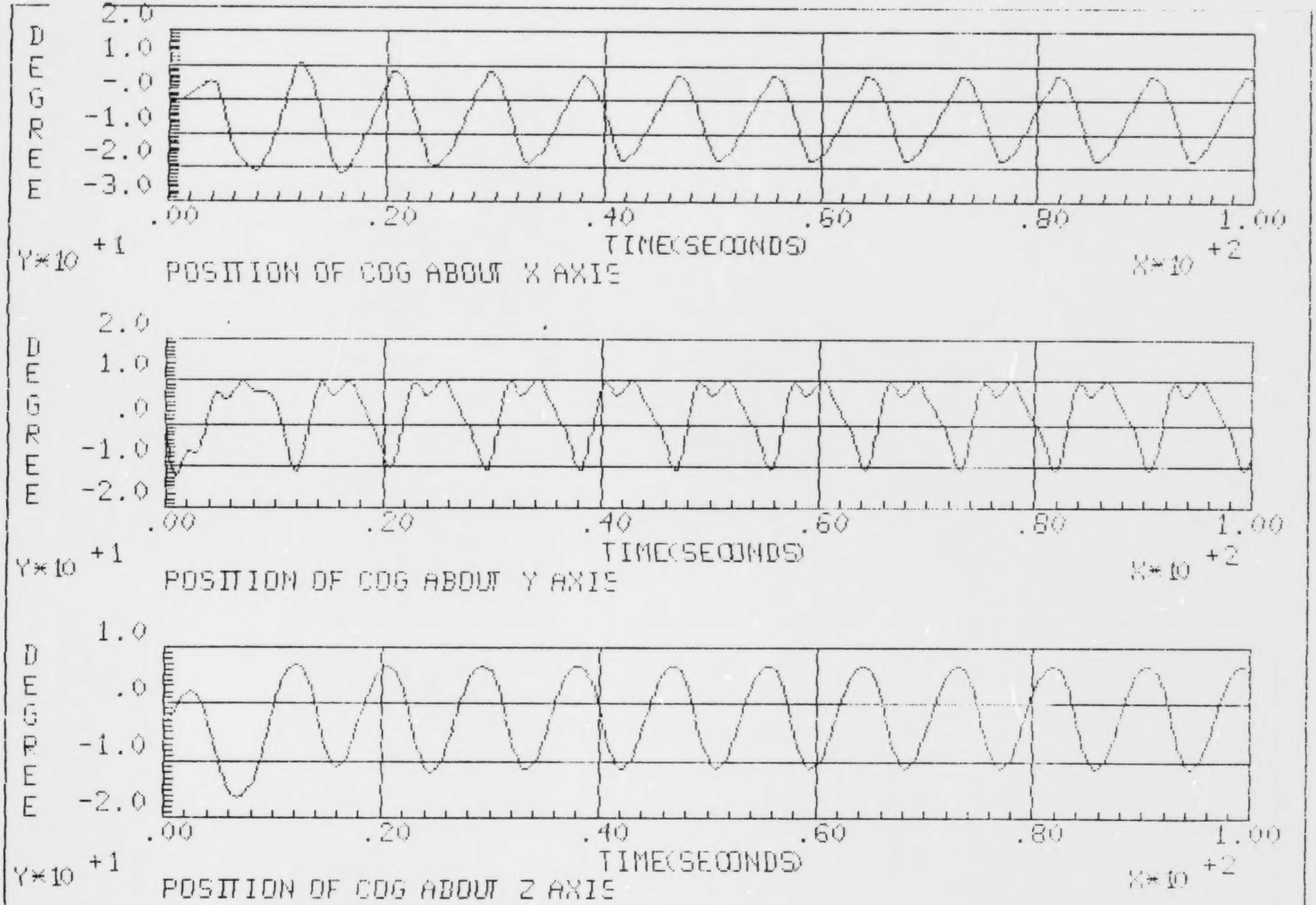
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

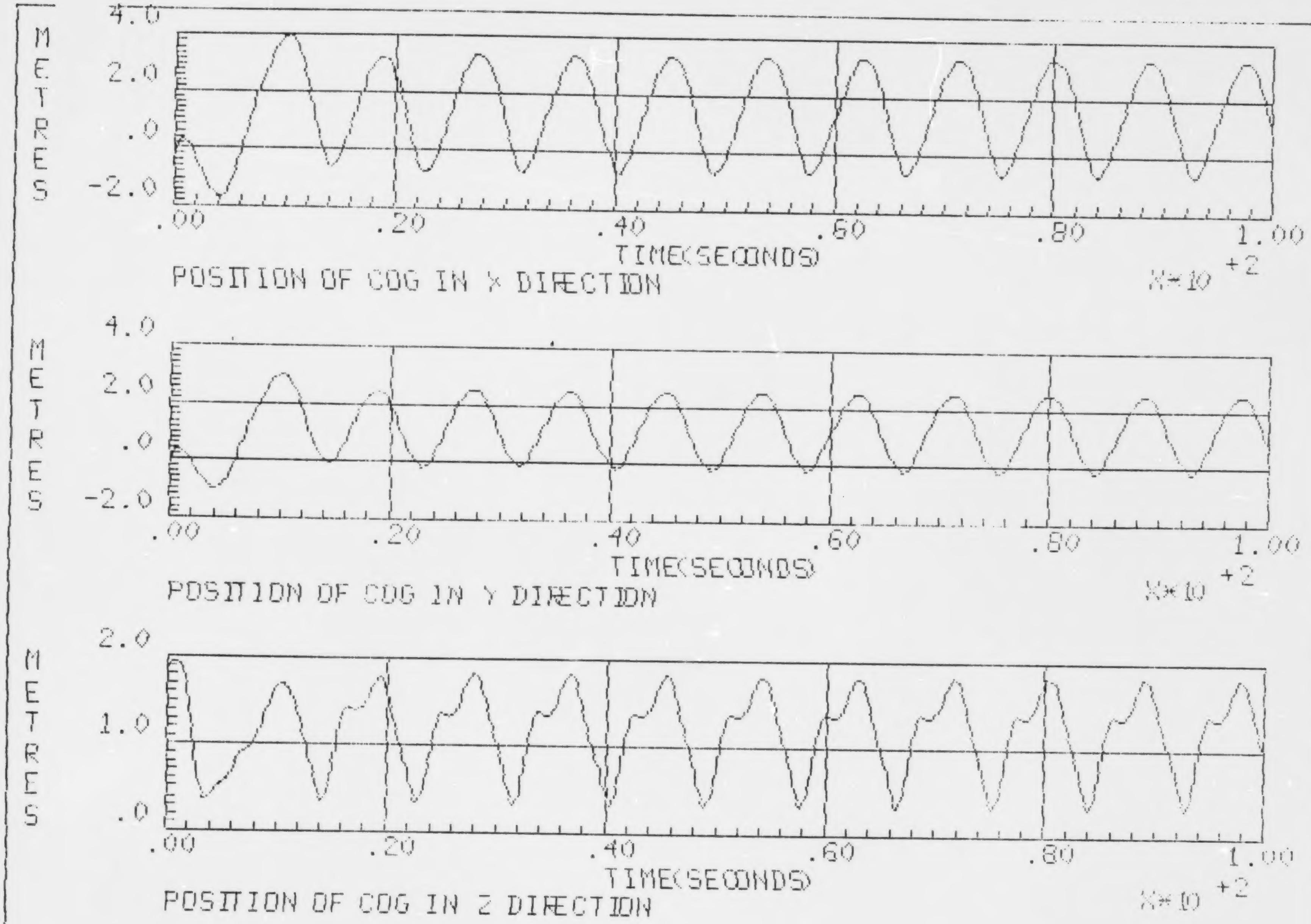
ROUND BILGE VESSEL CASE NO. ESM5-10%

Te=Tr, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=10.0% CRIT., Servo 2



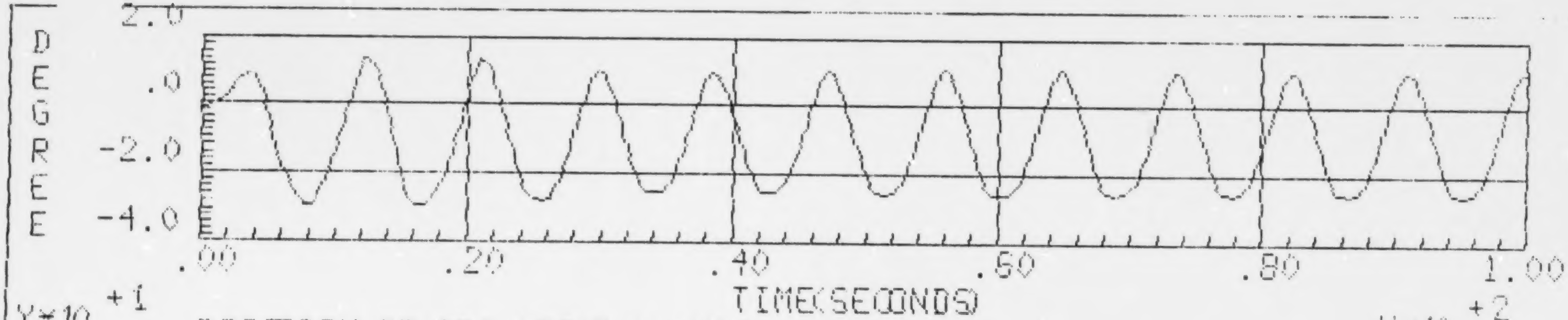
ROUND BILGE VESSEL CASE NO. ESM5-10%

Te=Tr, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=10.0% CRIT., Servo 2

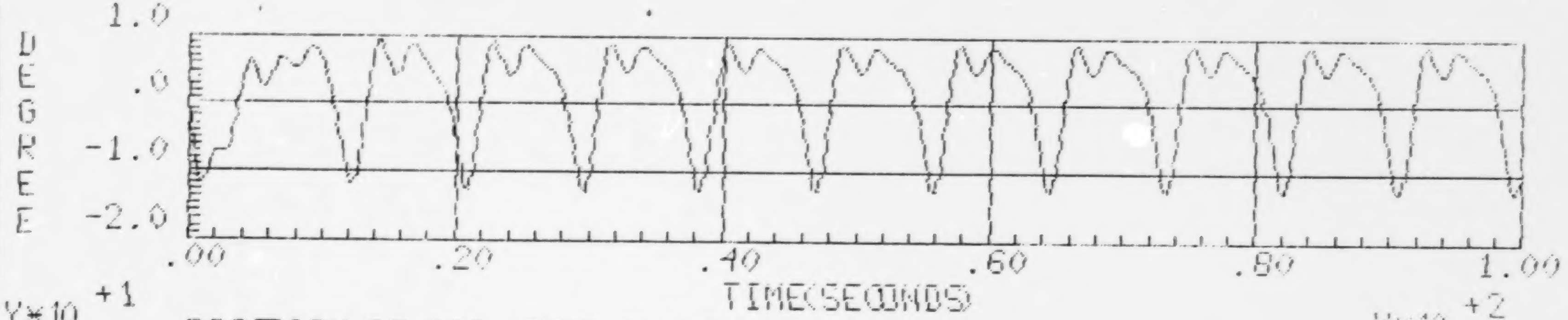


ROUND CHINE VESSEL CASE NO. ESM6-10%

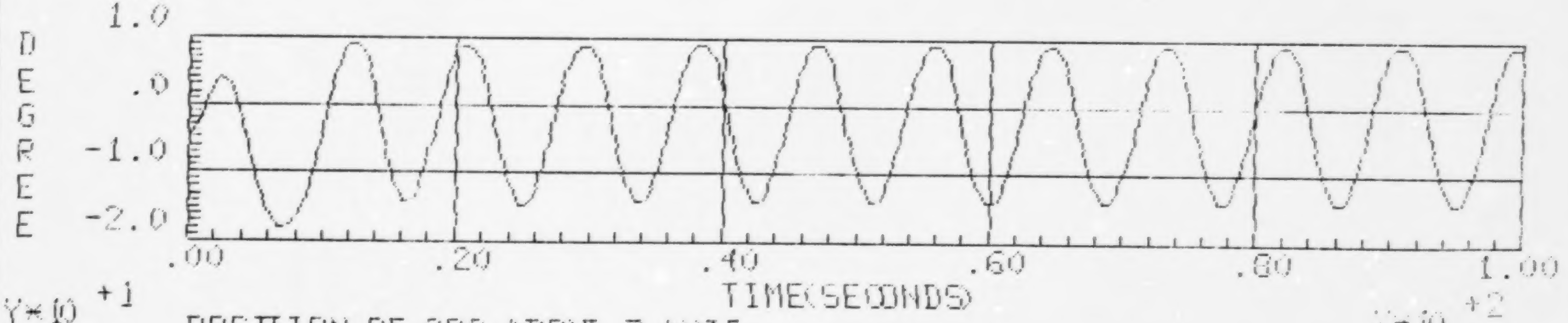
Te=Tr, Dir.=50 °, H=3.24 m, Diff.=0, D_{44}=10.0% CRIT., Servo 2



POSITION OF COG ABOUT X AXIS



POSITION OF COG ABOUT Y AXIS



POSITION OF COG ABOUT Z AXIS

ROUND CHINE VESSEL CASE NO. ESM6-10%

Te=Tr, Dir.=50 °, H=3.24 m, Diff.=0, D_{44}=10.0% CRIT., Servo 2

APPENDIX F

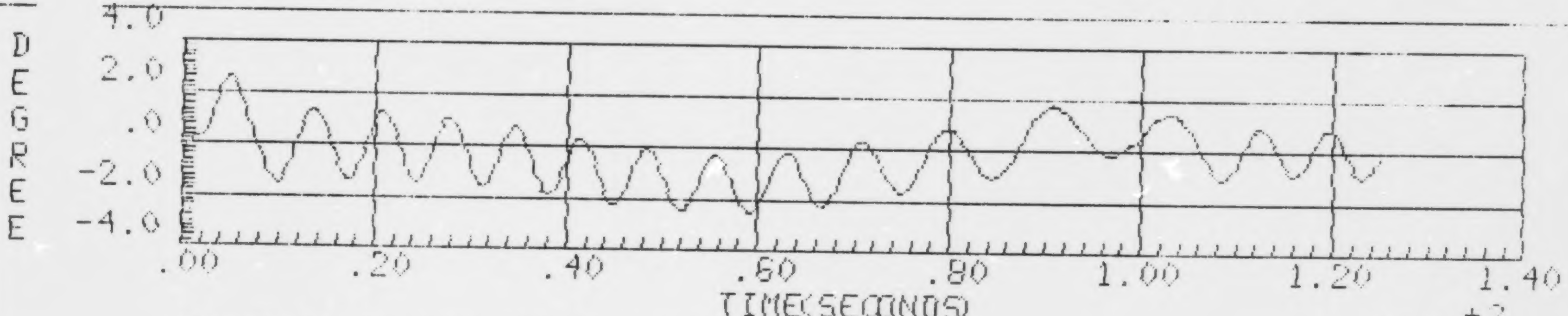
RESULTS OF "WAVE CREST" CASE

HARD CHINE VESSEL

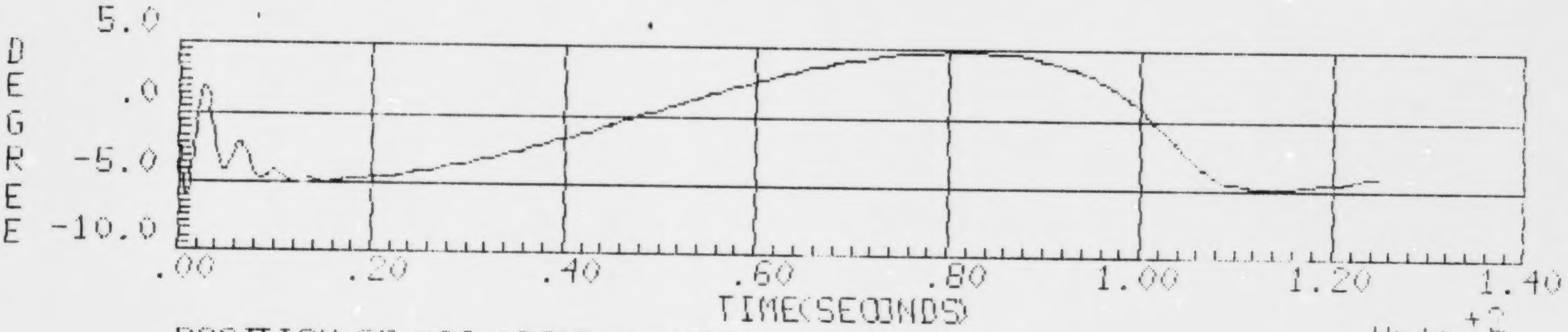
Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
WLS1	5.0	1/12	0.4
WLS2	30.0	1/12	0.4
WLS3	50.0	1/12	0.4
WLS4	5.0	1/7	0.4
WLS5	30.0	1/7	0.4
WLS6	50.0	1/7	0.4
WLS4-10%	5.0	1/7	10.0
WLS5-10%	30.0	1/7	10.0

ROUND BILGE VESSEL

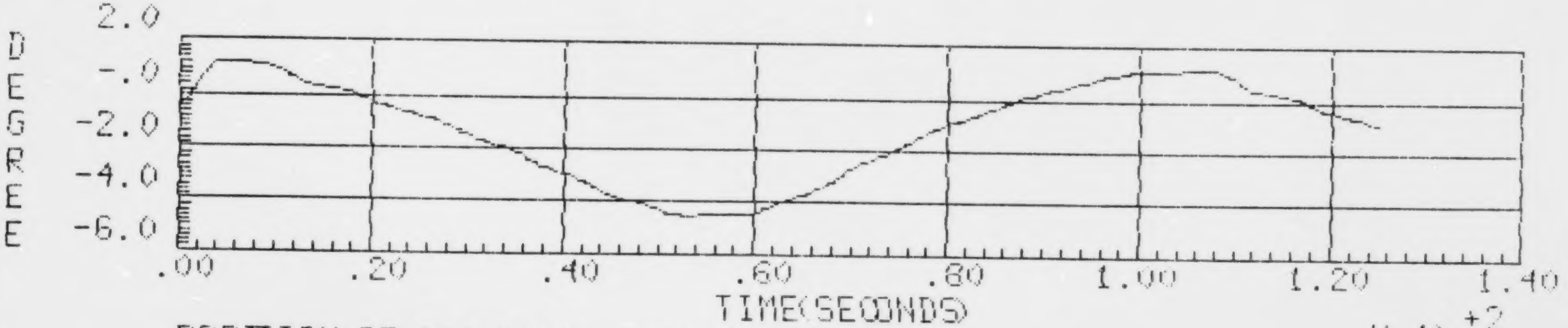
Case number	wave direction	wave height (fraction of wave length)	damping (% critical)
ELS4	5.0	1/7	0.4
ELS5	30.0	1/7	0.4
ELS6	50.0	1/7	0.4



POSITION OF COG ABOUT X AXIS



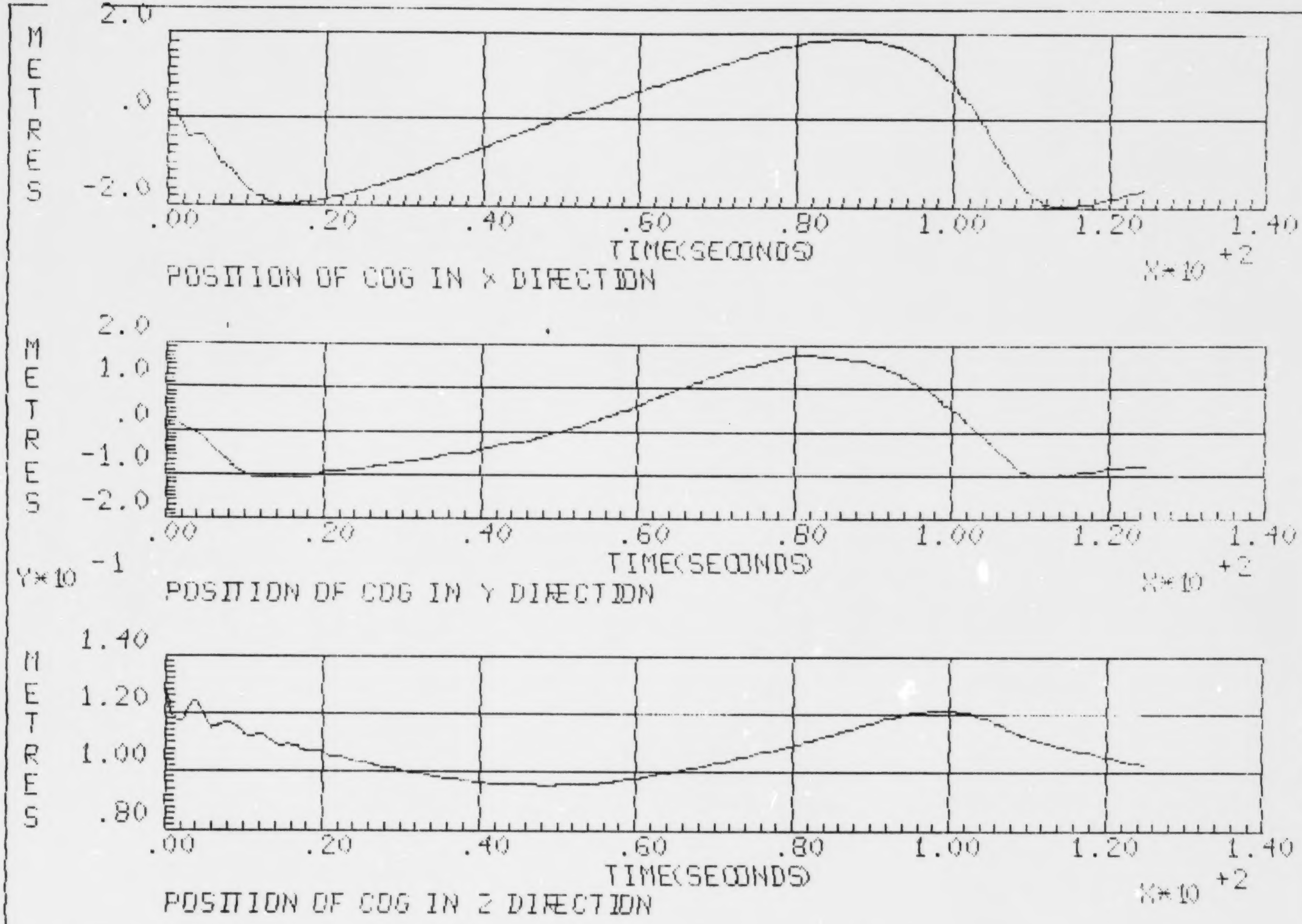
POSITION OF COG ABOUT Y AXIS



POSITION OF COG ABOUT Z AXIS

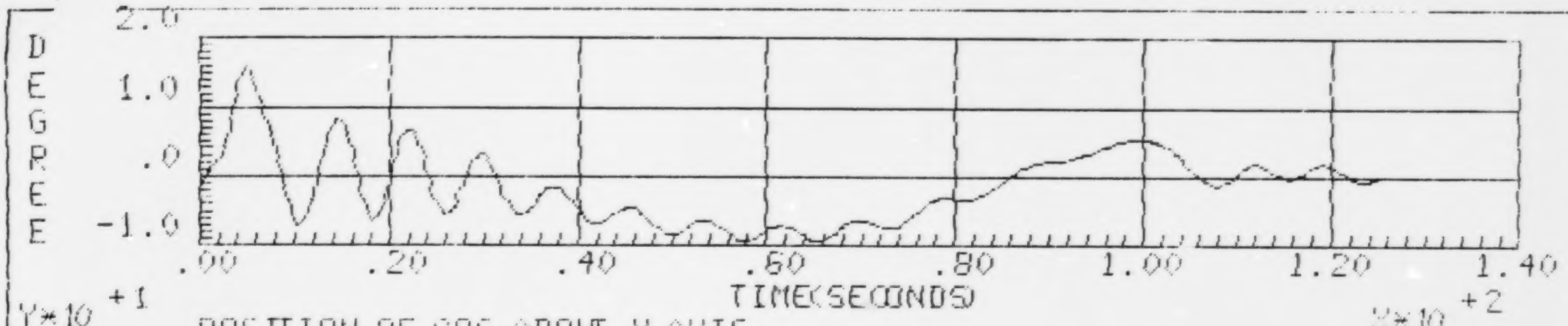
HARD CHINE VESSEL CASE NO. WLS1

Te=100 s, Dir.=5 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2

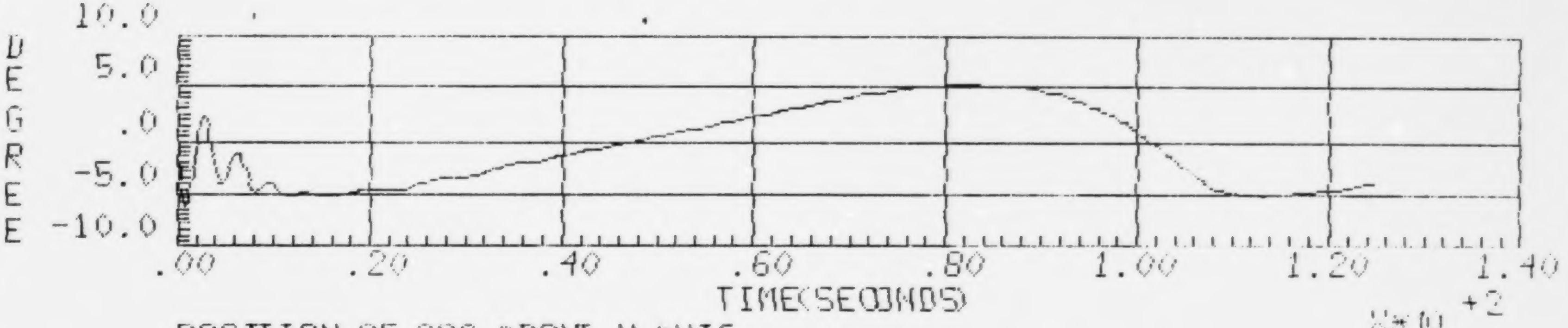


HARD CHINE VESSEL CASE NO. WLS1

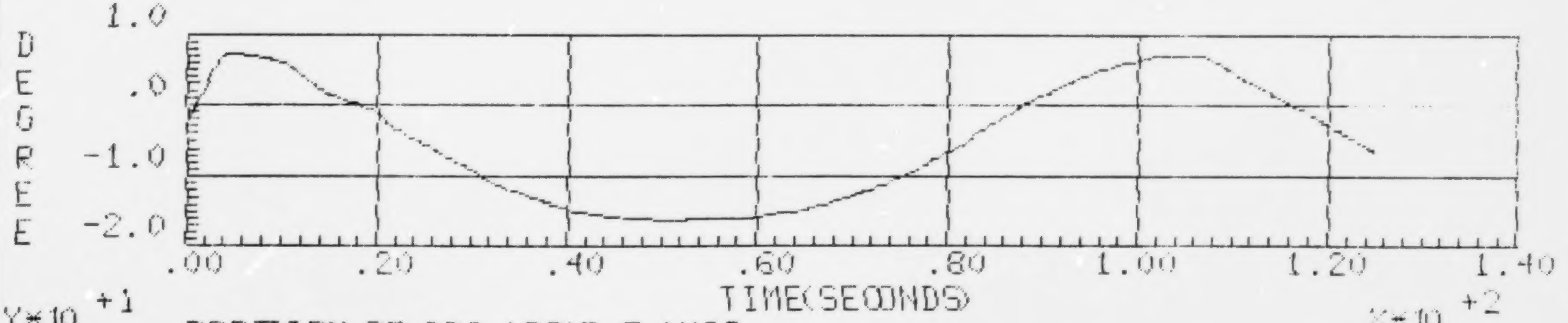
Te=100 s, Dir.=5 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG ABOUT X AXIS



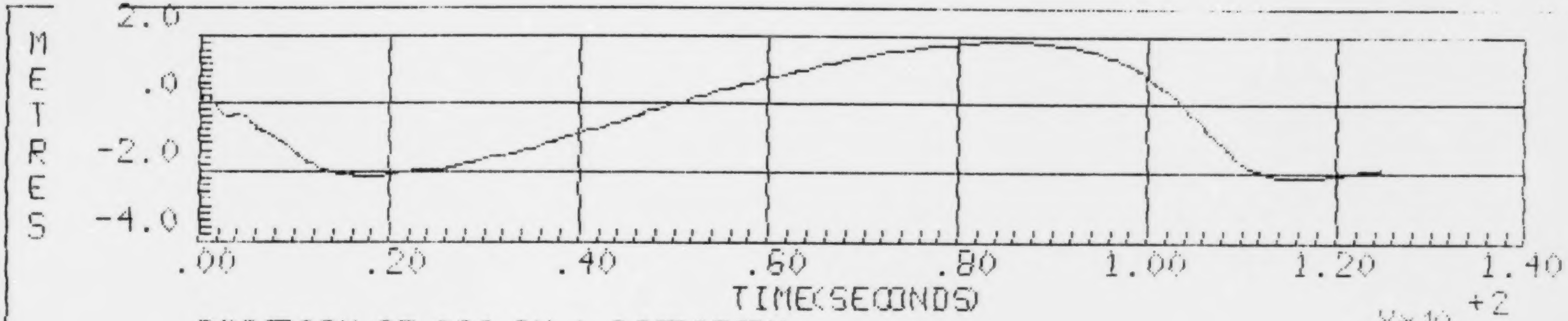
POSITION OF COG ABOUT Y AXIS



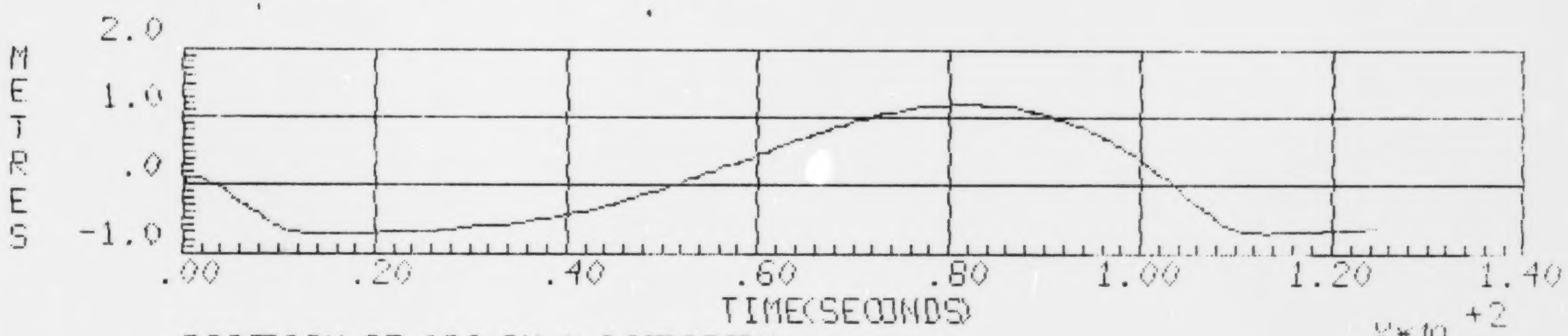
POSITION OF COG ABOUT Z AXIS

HARD CHINE VESSEL CASE NO. WLS2

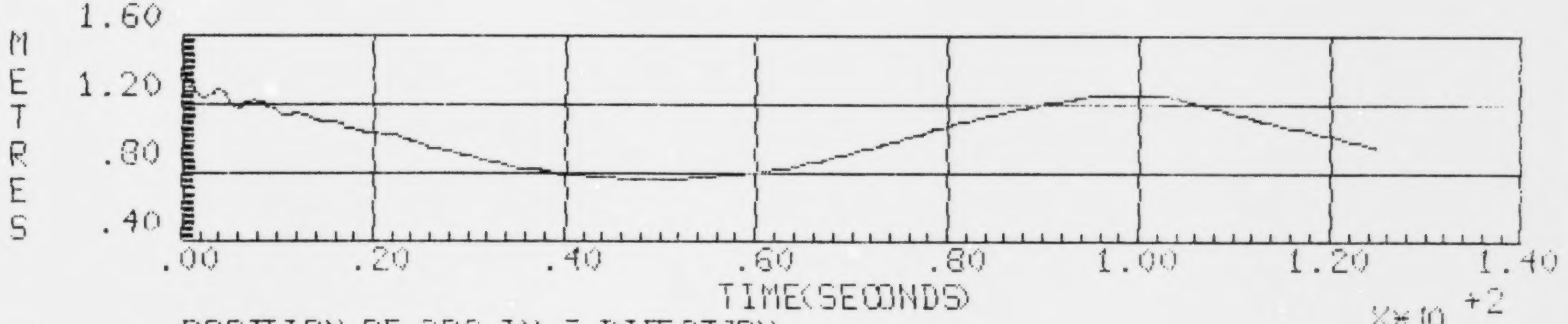
Te=100 s, Dir.=30 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



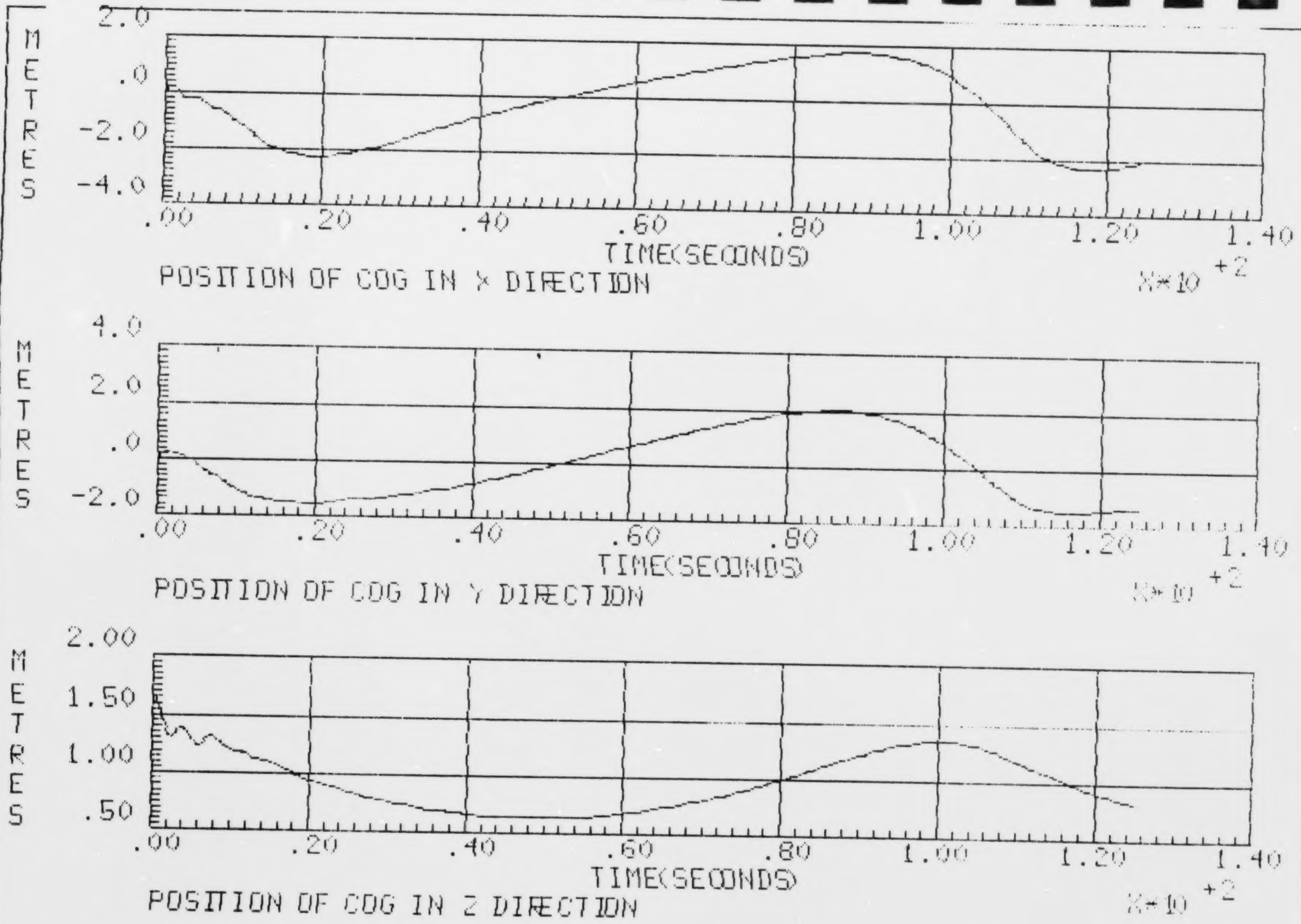
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

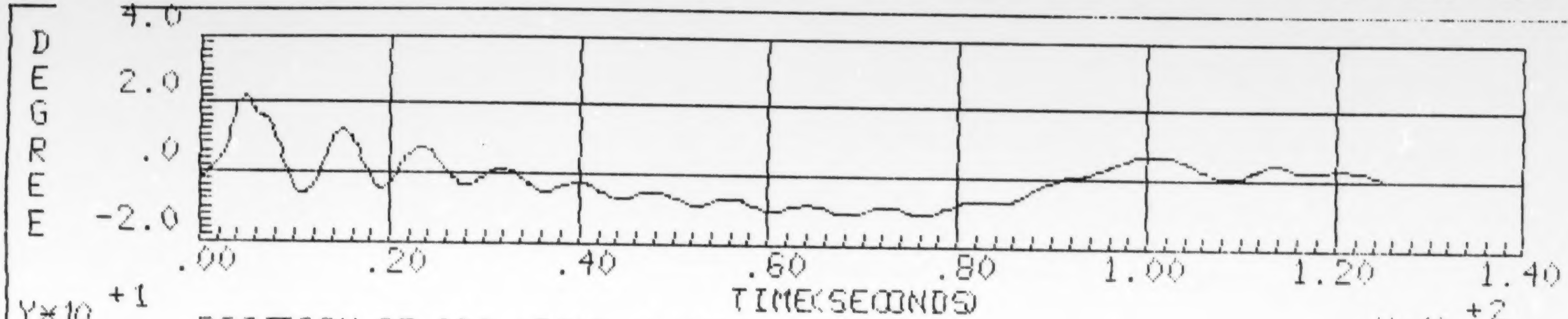
HARD CHINE VESSEL CASE NO. WLS2

Te=100 s, Dir.=30 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2

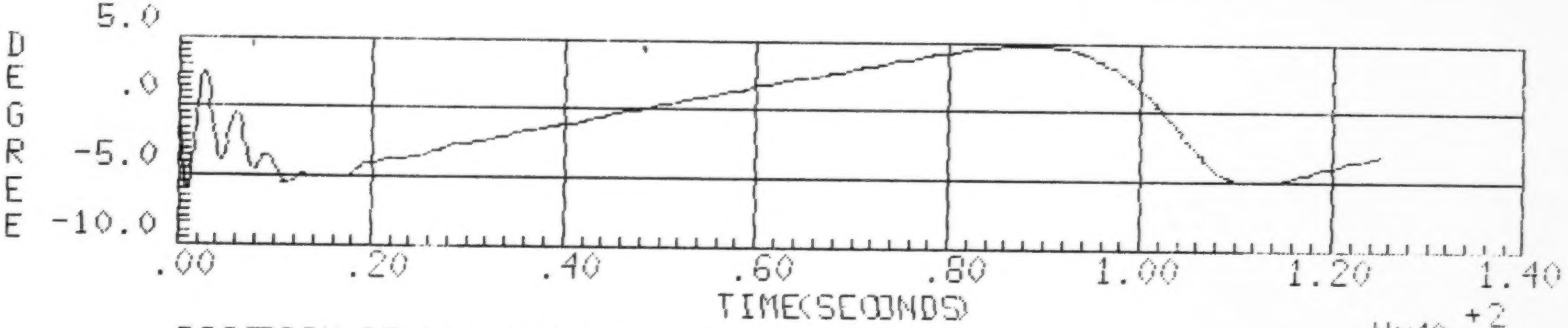


HARD CHINE VESSEL. CASE NO. WLS3

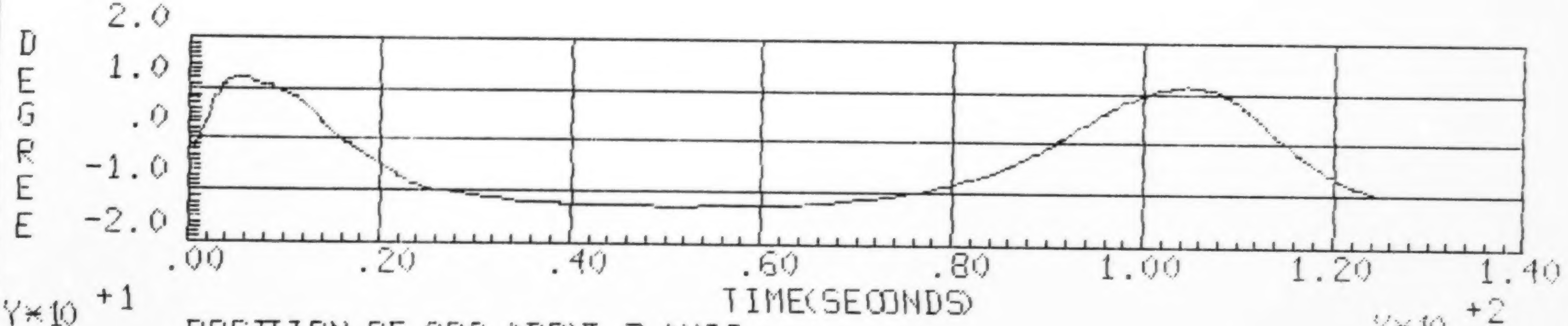
Te=100 s, Dir.=50 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG ABOUT X AXIS



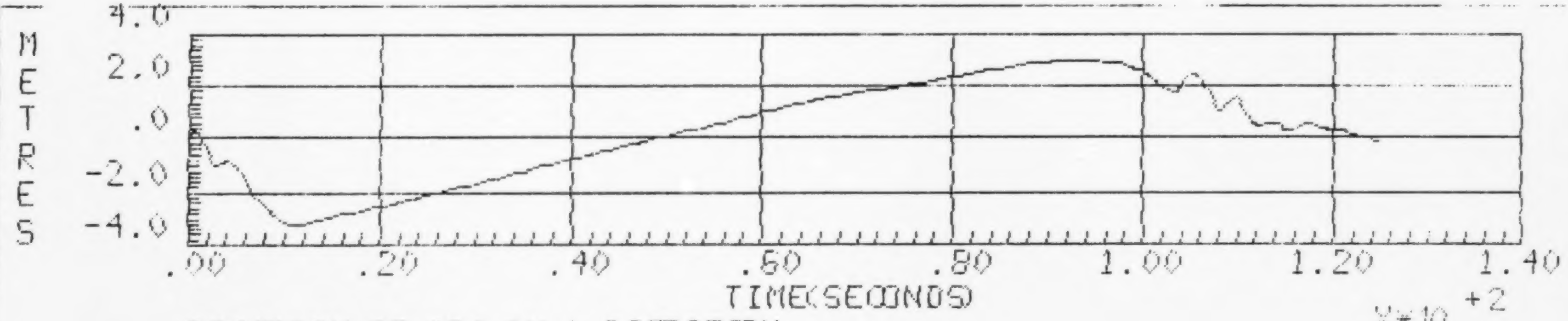
POSITION OF COG ABOUT Y AXIS



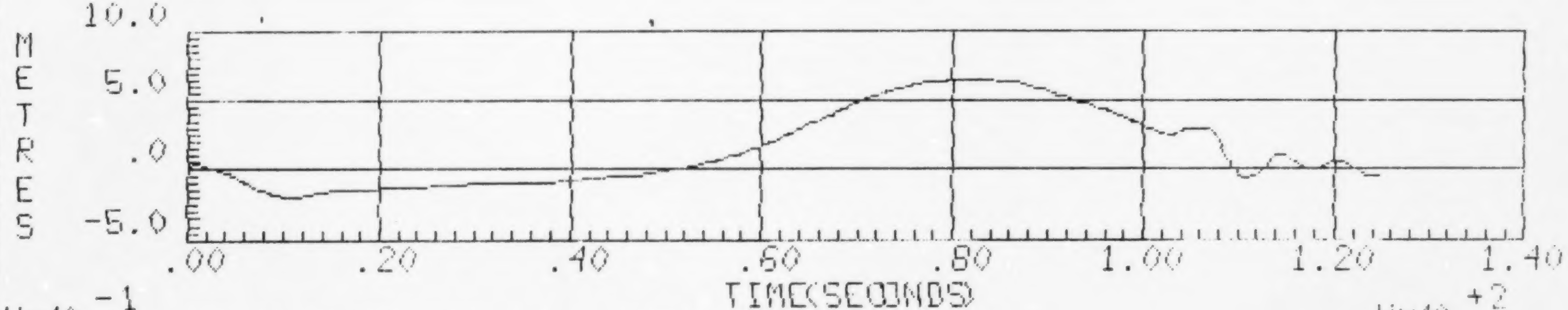
POSITION OF COG ABOUT Z AXIS

HARD CHINE VESSEL CASE NO. WLS3

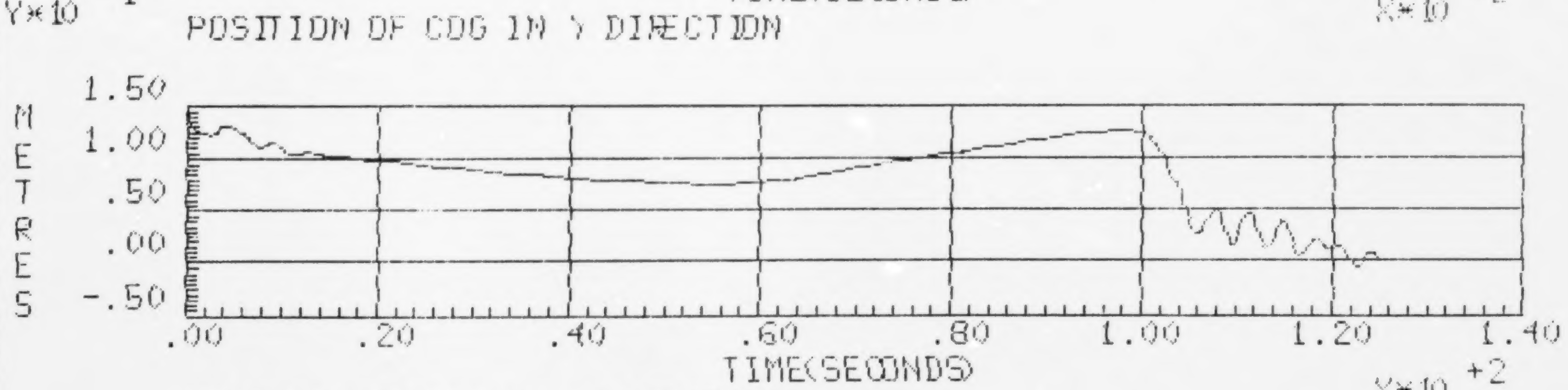
Te=100 s, Dir.=50 °, H=1.91 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



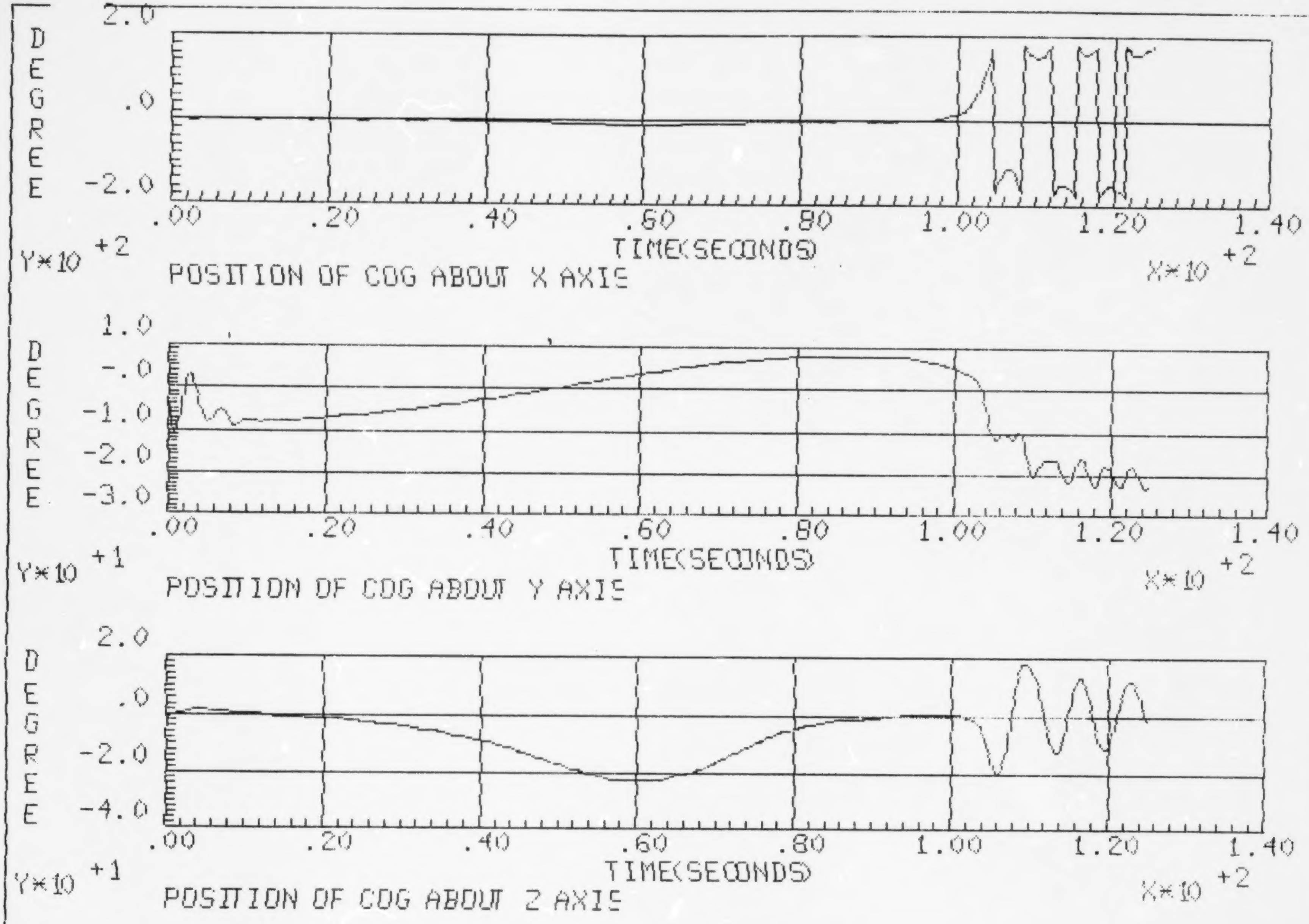
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

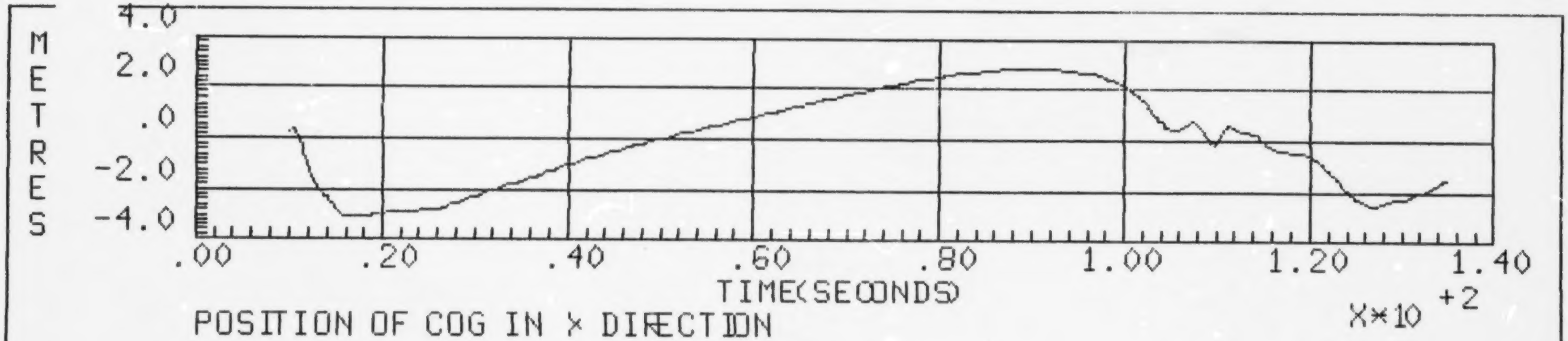
HARD CHINE VESSEL CASE NO. WLS4

Te=100 s, Dir.=5 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2

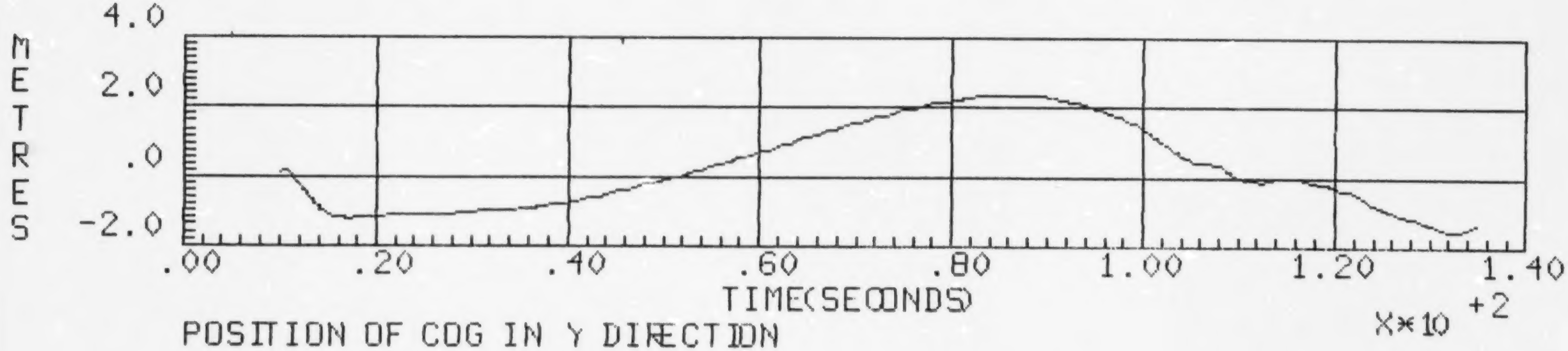


HARD CHINE VESSEL CASE NO. WLS4

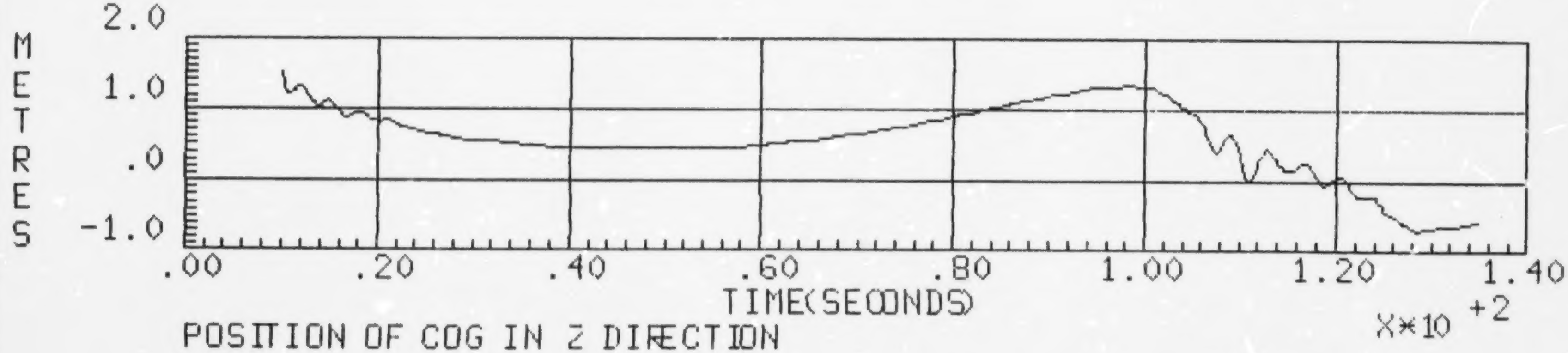
Te=100 s, Dir.=5 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



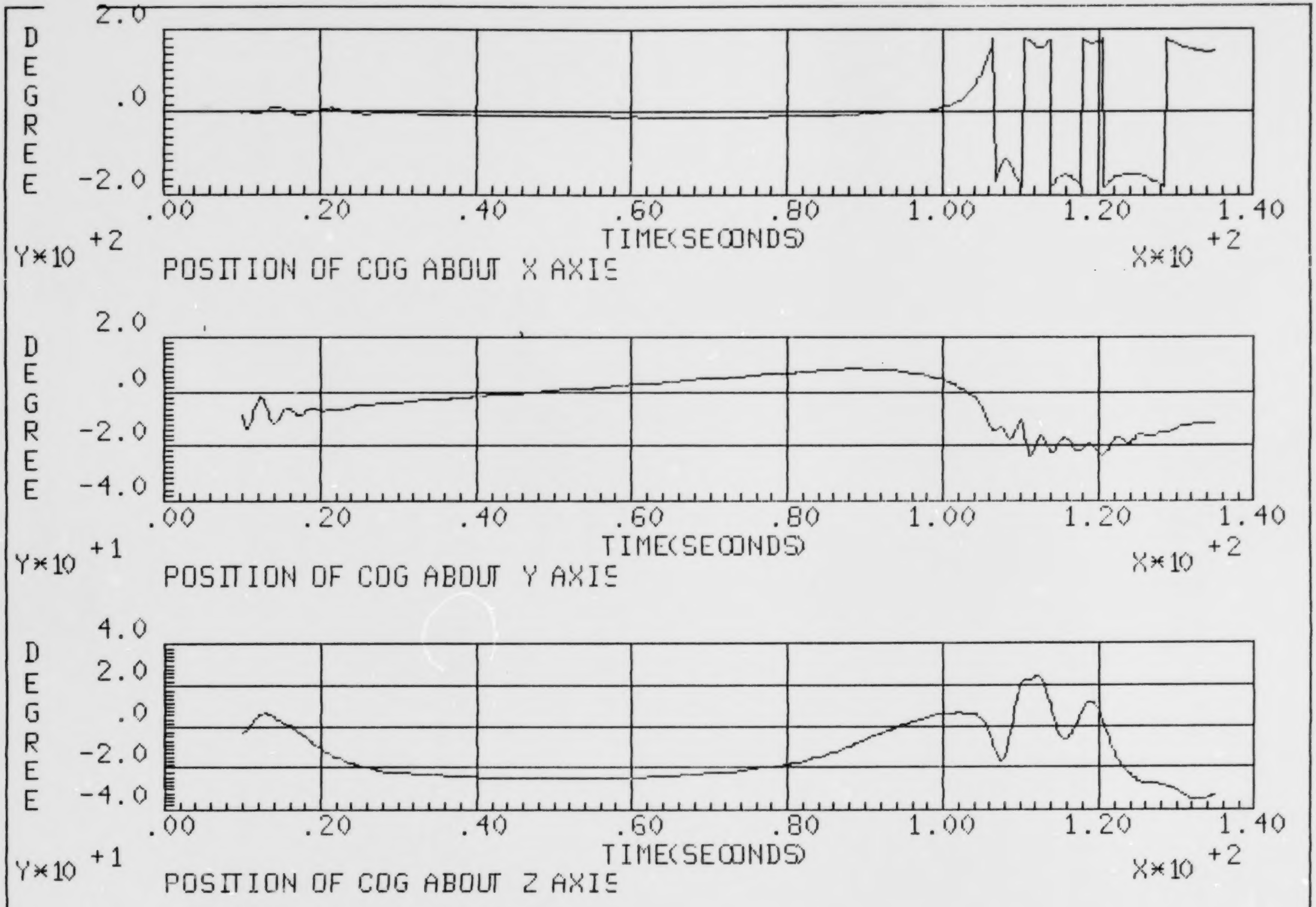
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

HARD CHINE VESSEL CASE NO. WLS5

Te=100 s, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



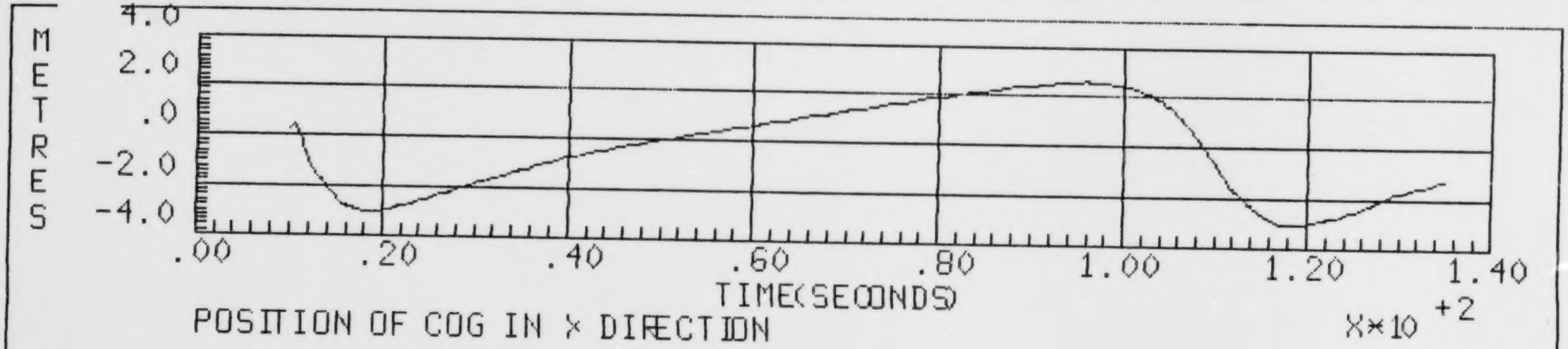
POSITION OF COG ABOUT X AXIS

POSITION OF COG ABOUT Y AXIS

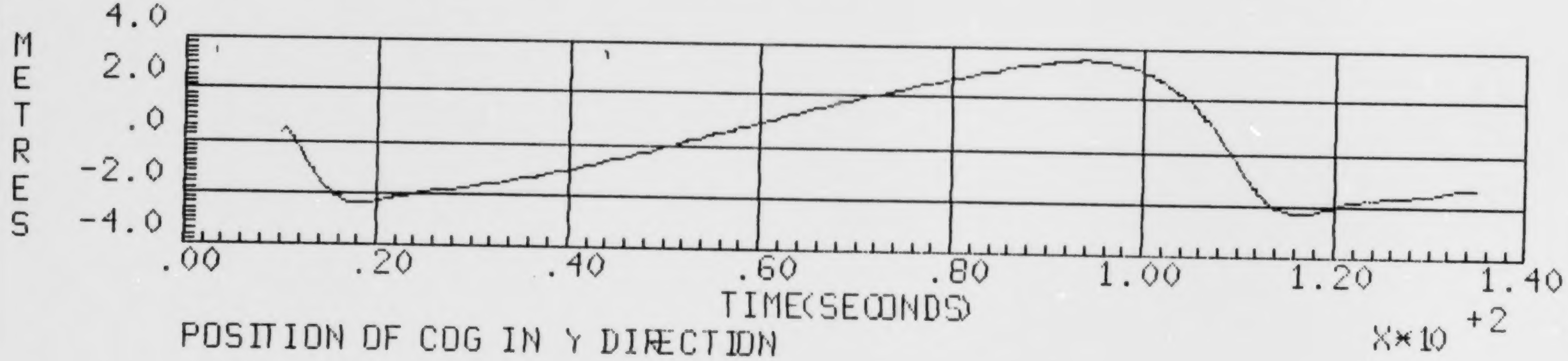
POSITION OF COG ABOUT Z AXIS

HARD CHINE VESSEL CASE NO. WLS5

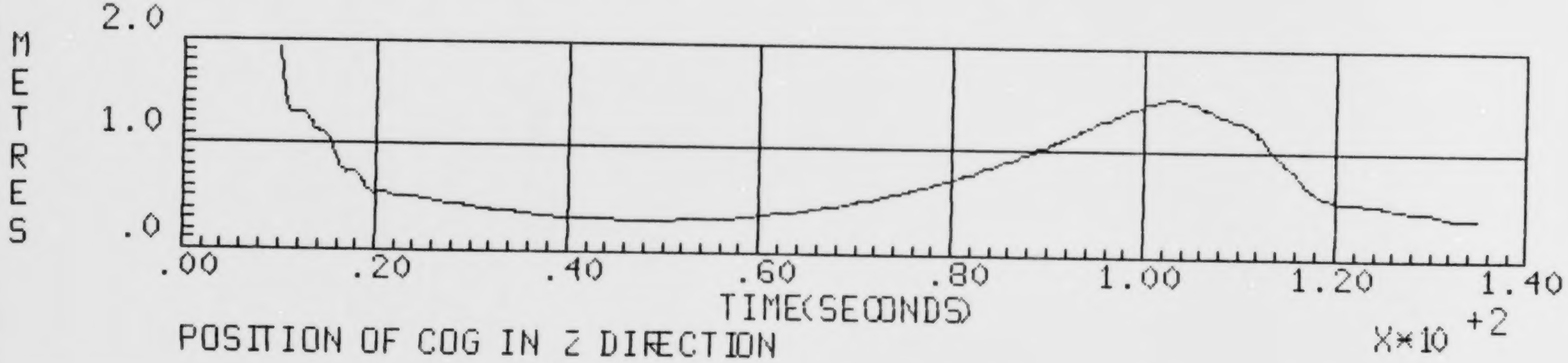
Te=100 s, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



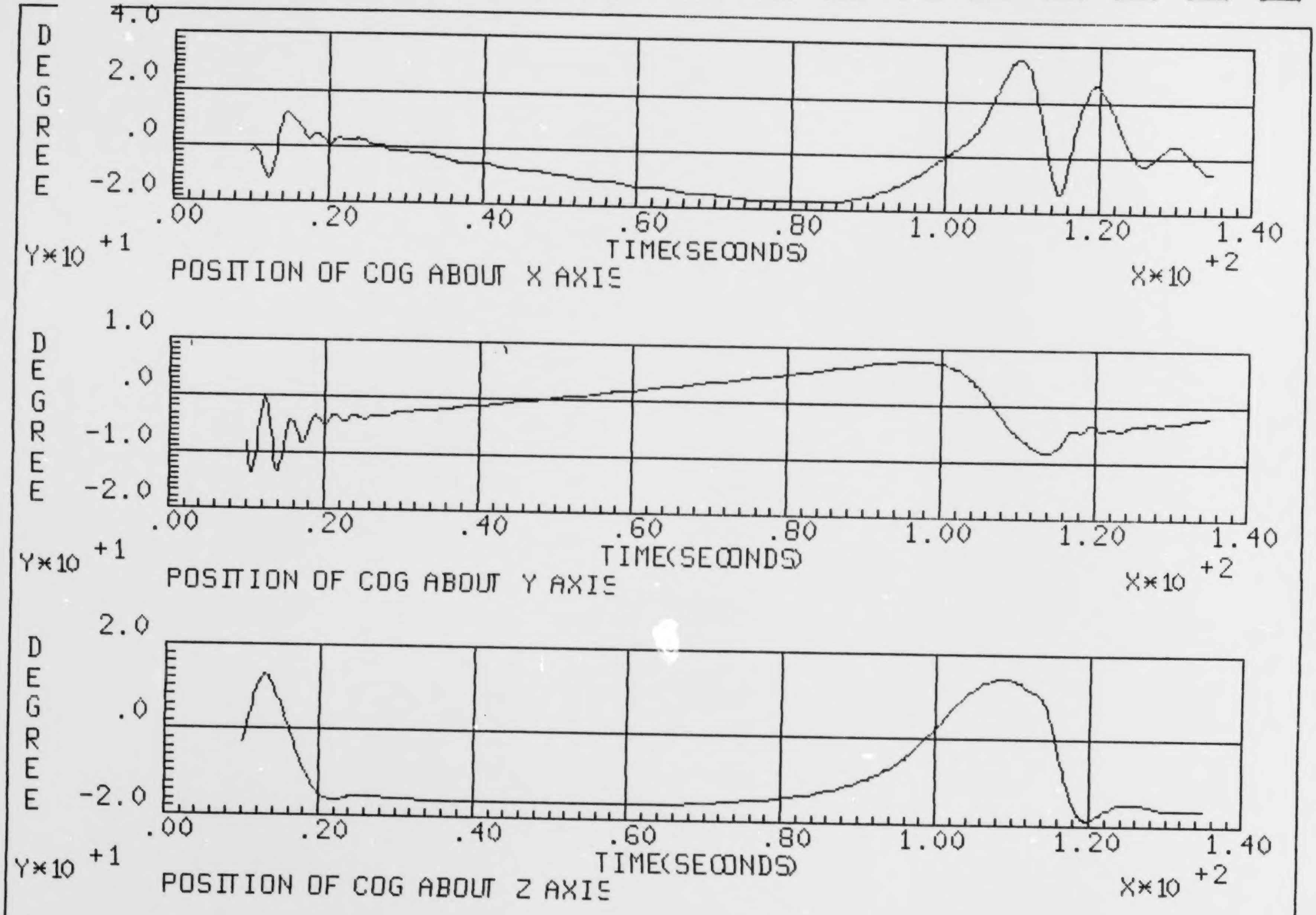
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

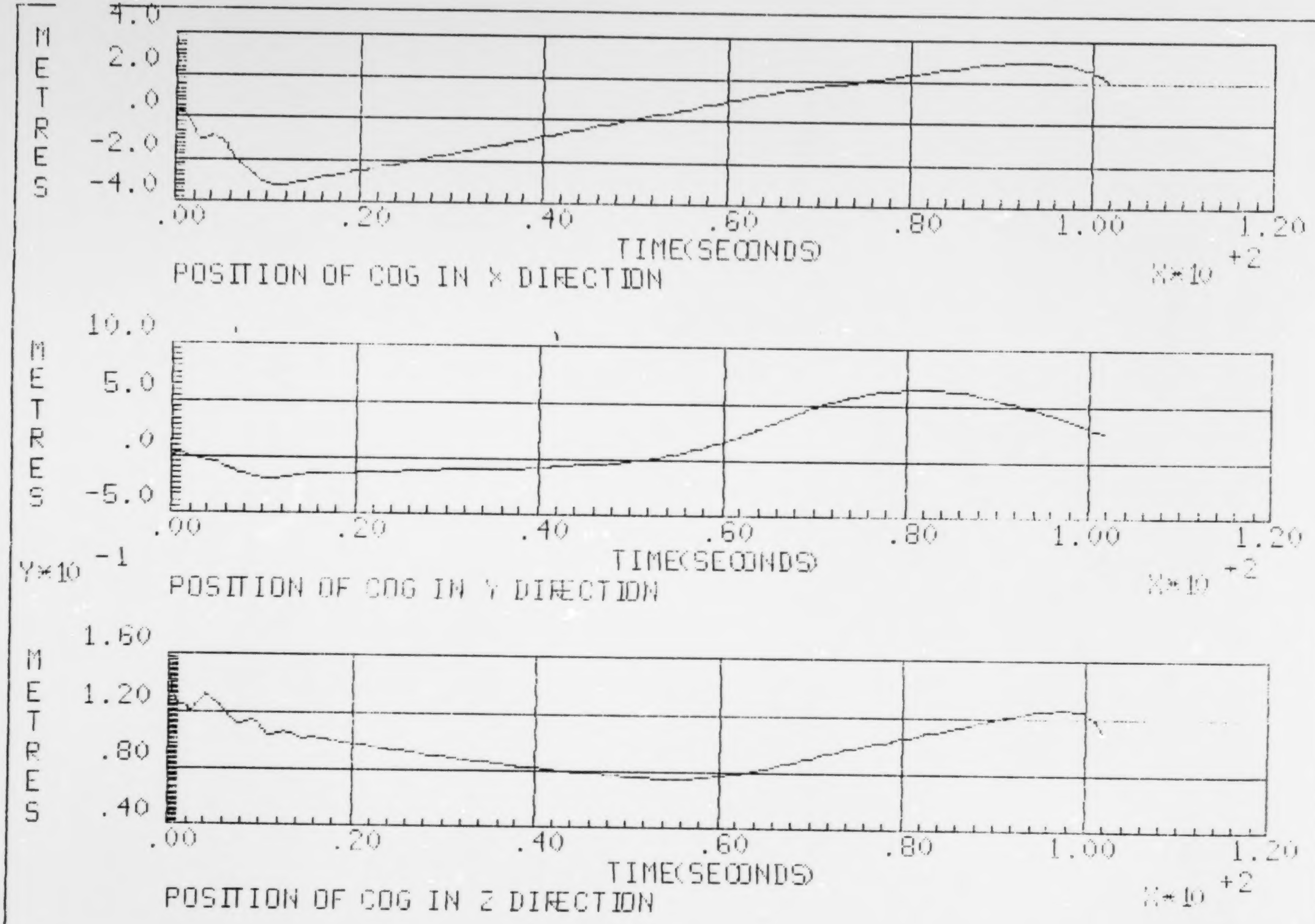
HARD CHINE VESSEL CASE NO. WLS6

Te=100 s, Dir.=50 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



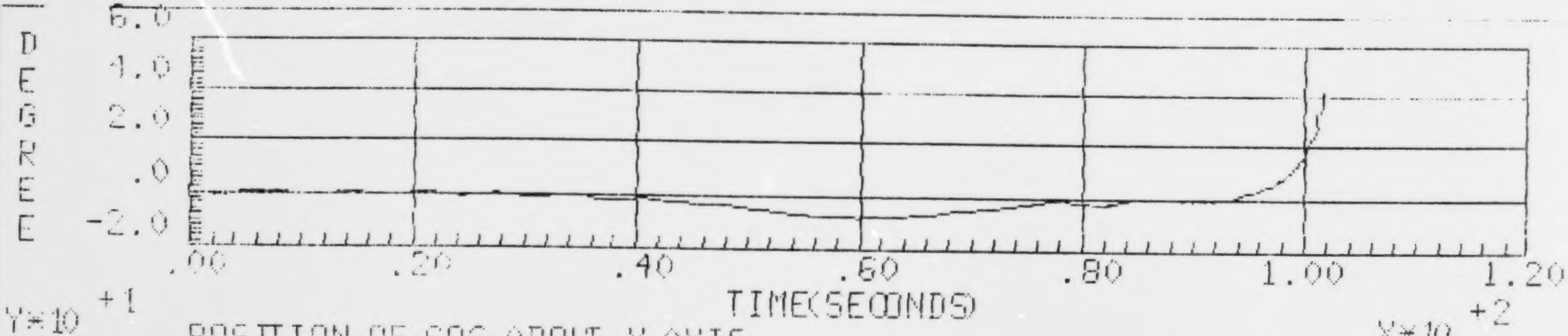
HARD CHINE VESSEL CASE NO. WLS6

Te=100 s, Dir.=50 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2

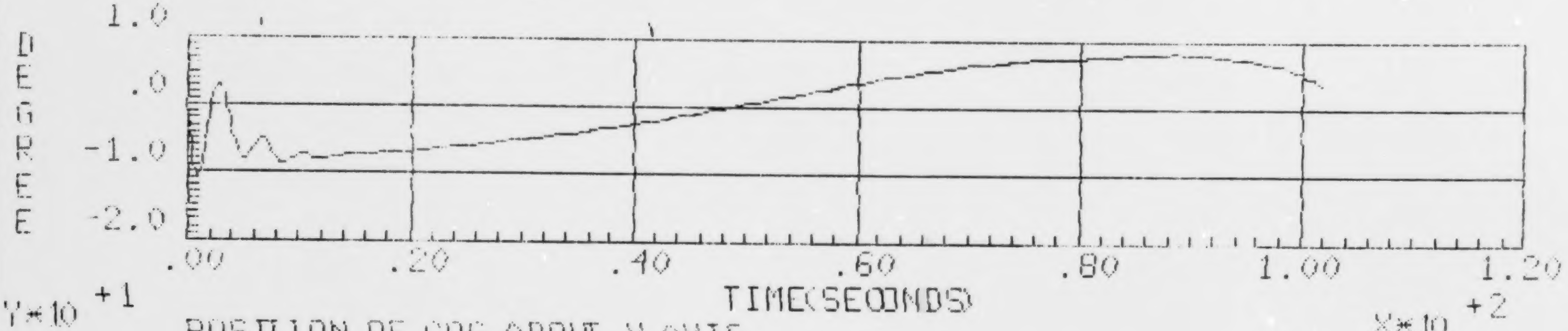


HARD CHINE VESSEL CASE NO. WLS4-10%

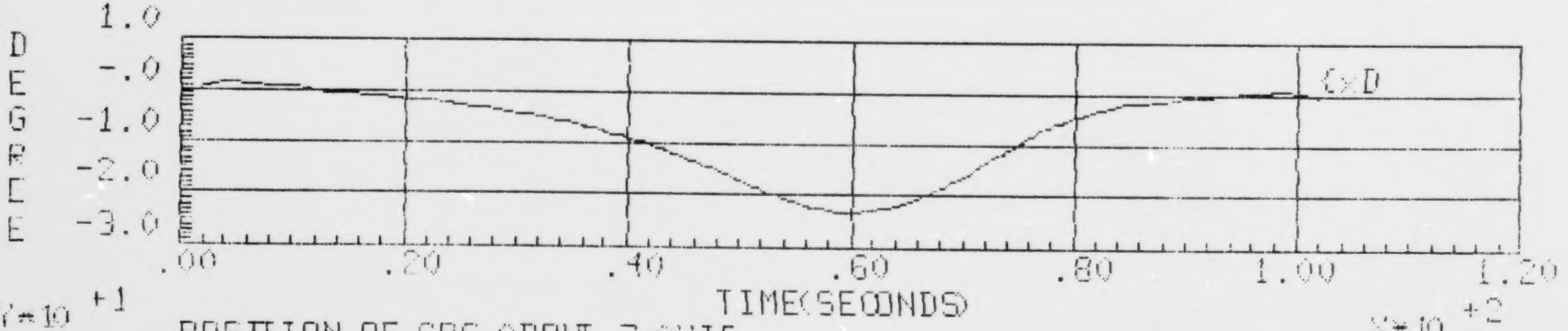
Te=100 s, Dir.=5 °, H=3.24 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



POSITION OF COG ABOUT X AXIS



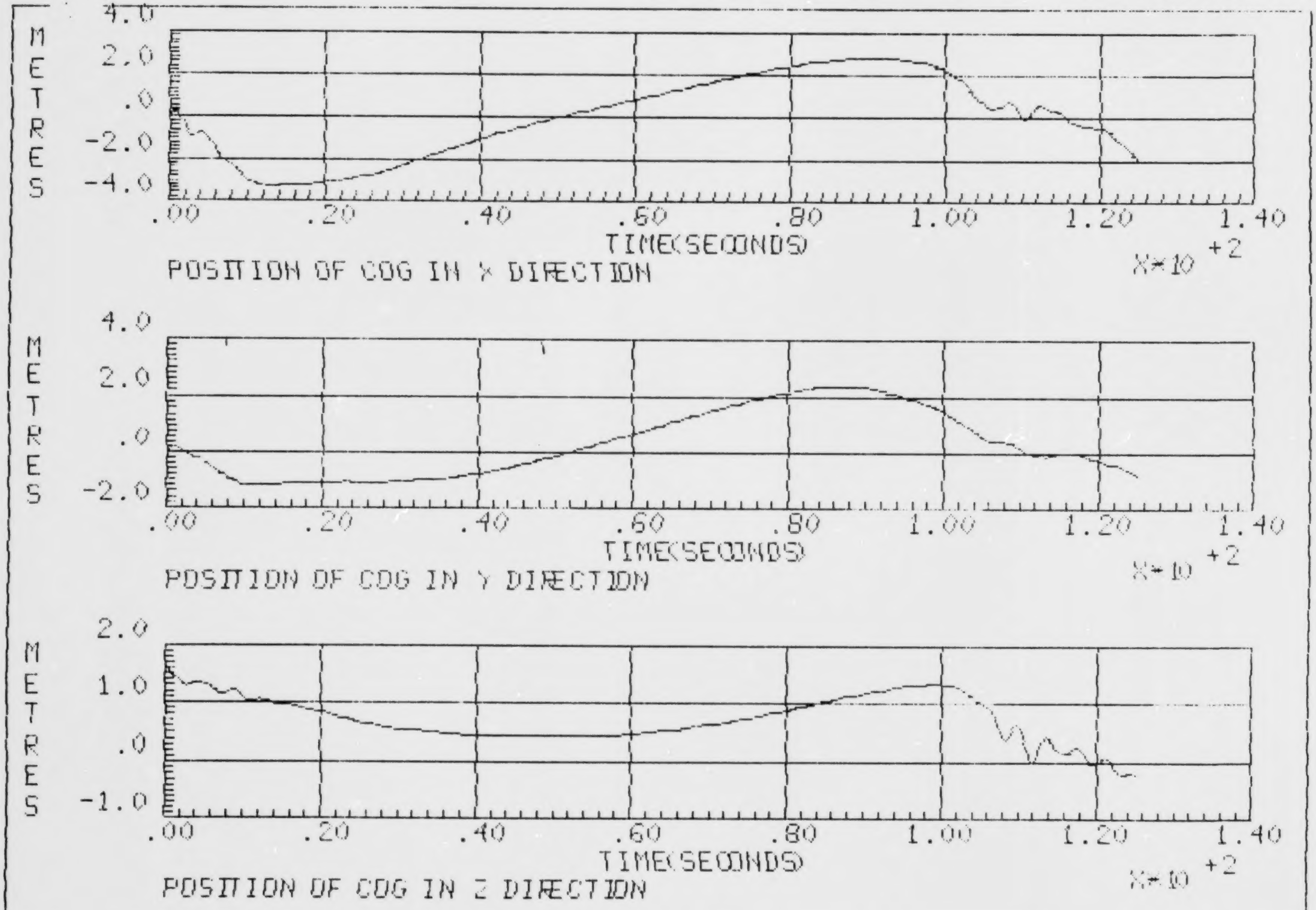
POSITION OF COG ABOUT Y AXIS



POSITION OF COG ABOUT Z AXIS

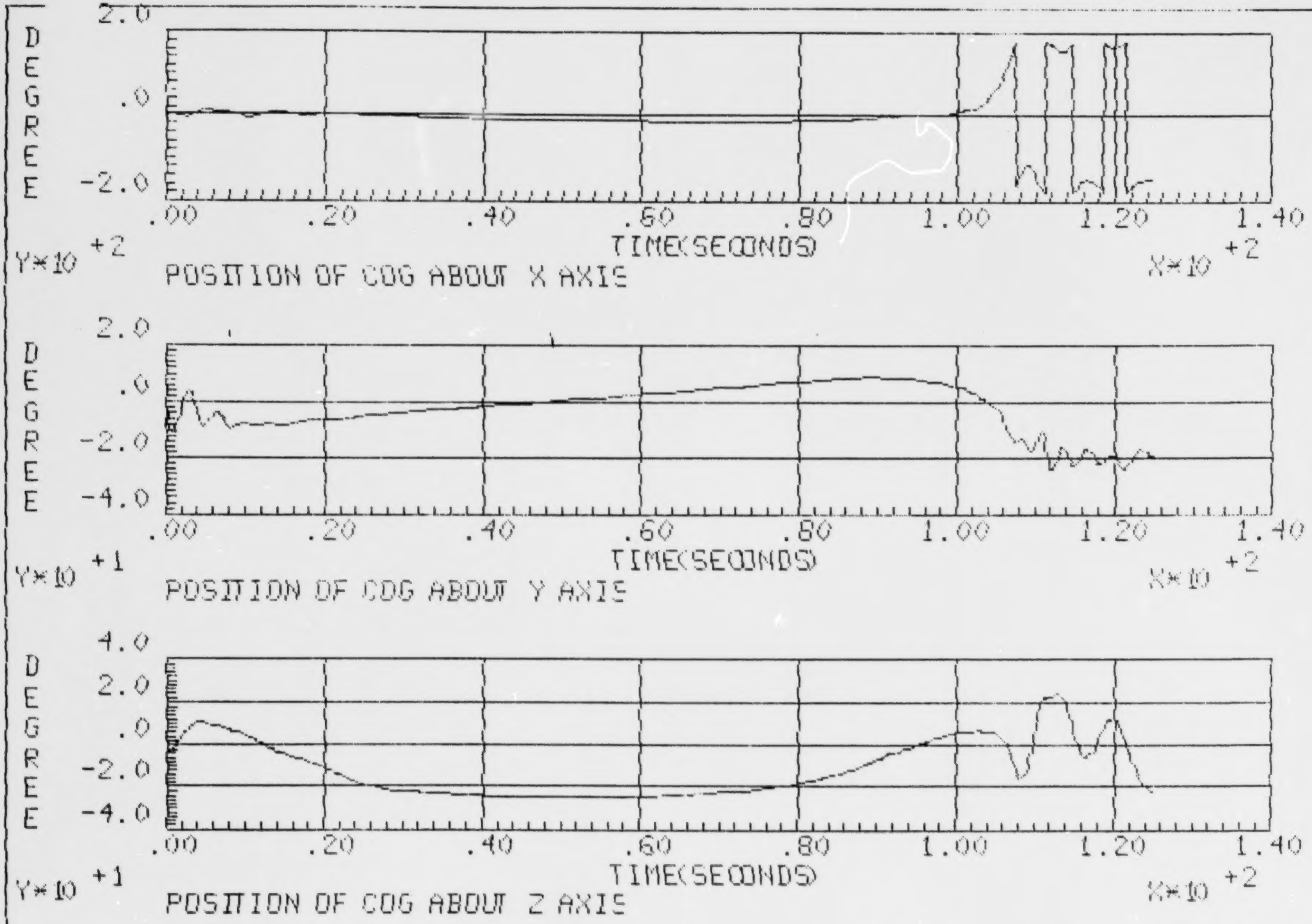
HARD CHINE VESSEL CASE NO. WLS4-10%

Te=100 s, Dir.=5 °, H=3.24 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



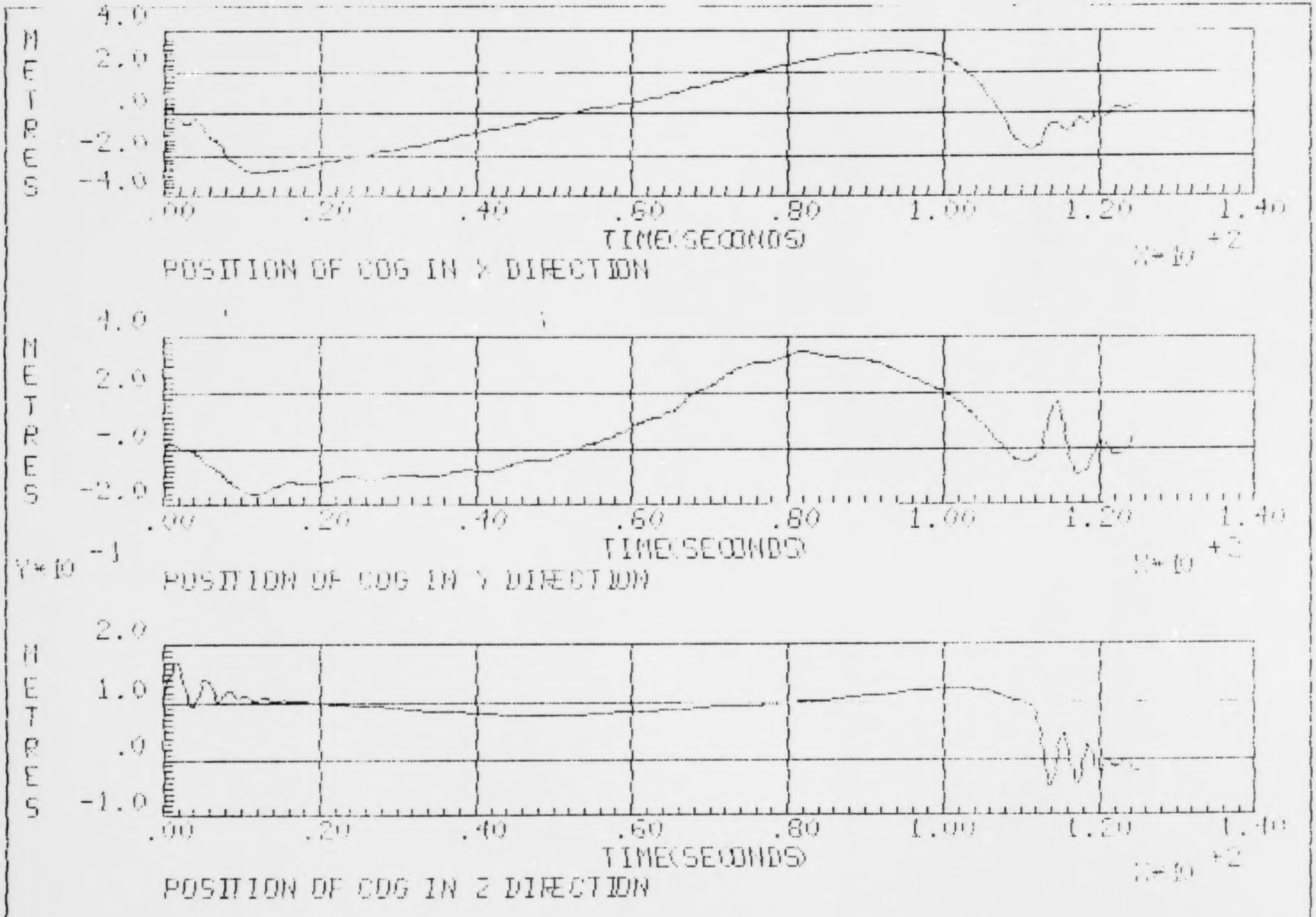
HARD CHINE VESSEL CASE NO. WLS5-10%

Te=100 s, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



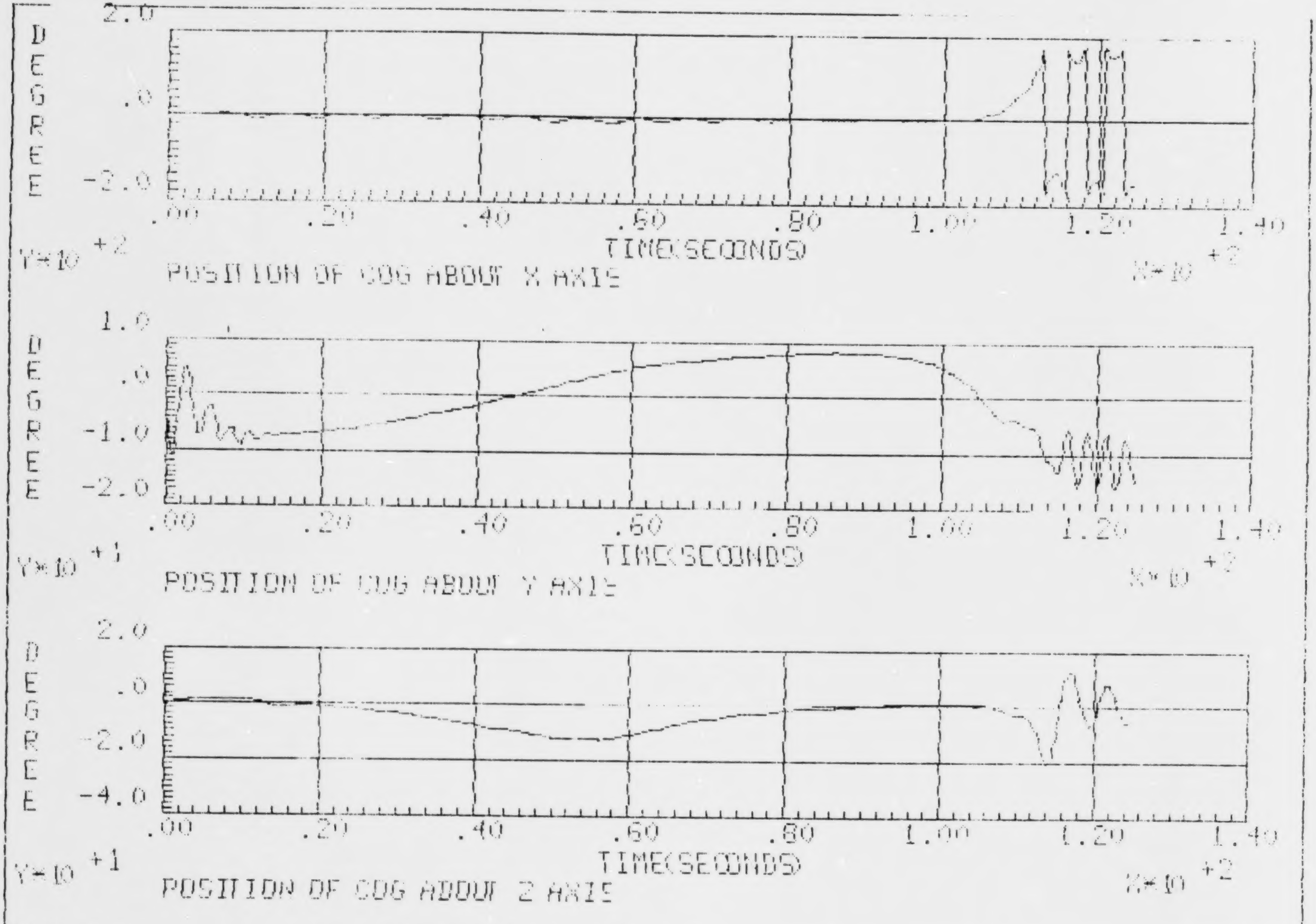
HARD CHINE VESSEL CASE NO. WLS5-10%

Te=100 s, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=10.0 % CRIT., Servo 2



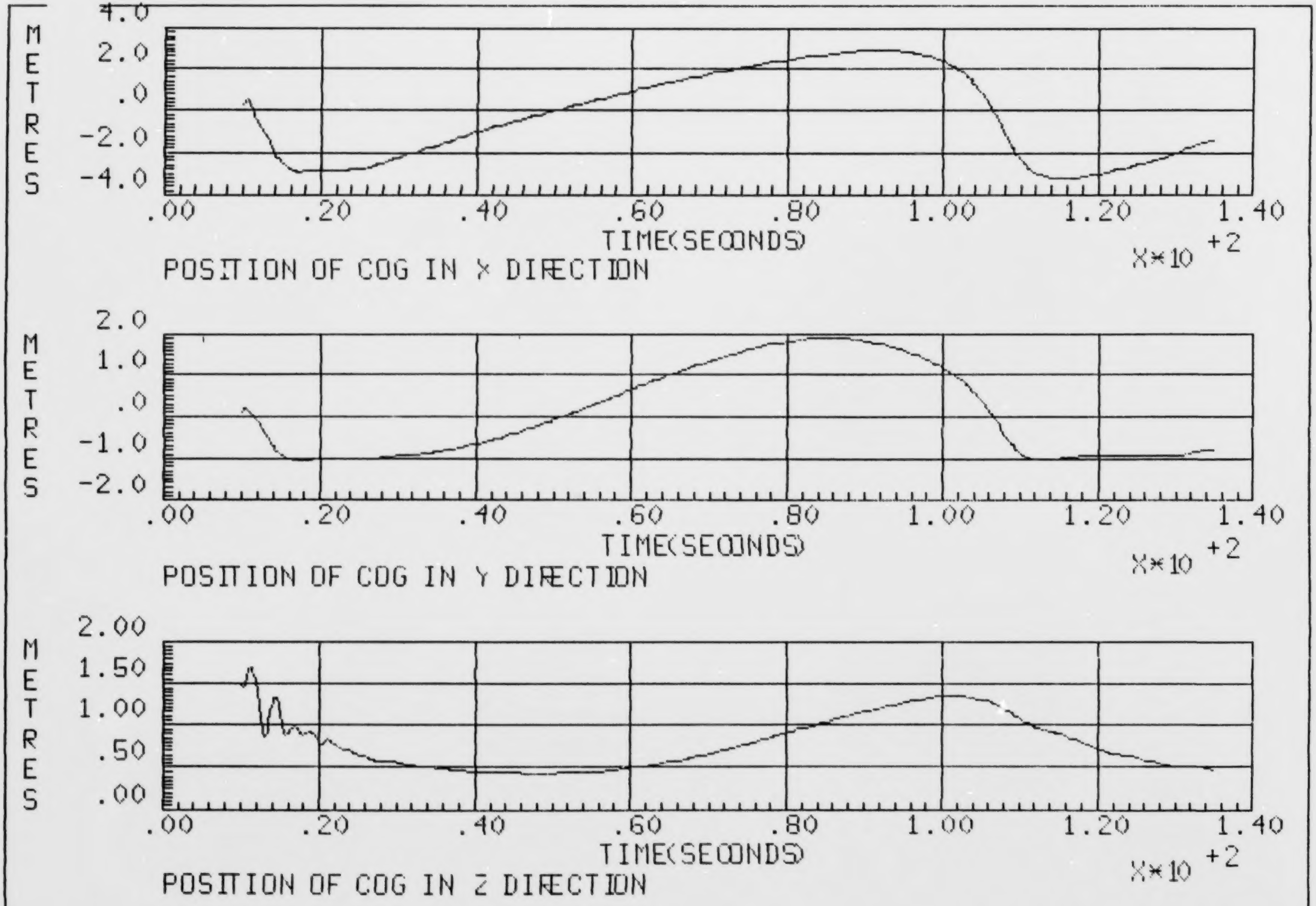
ROUND BILGE VESSEL CASE NO. ELS4

Te=100 s, Dir.=5 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



ROUND BILGE VESSEL CASE NO. ELS4

Te=100 s, Dir.=5 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2

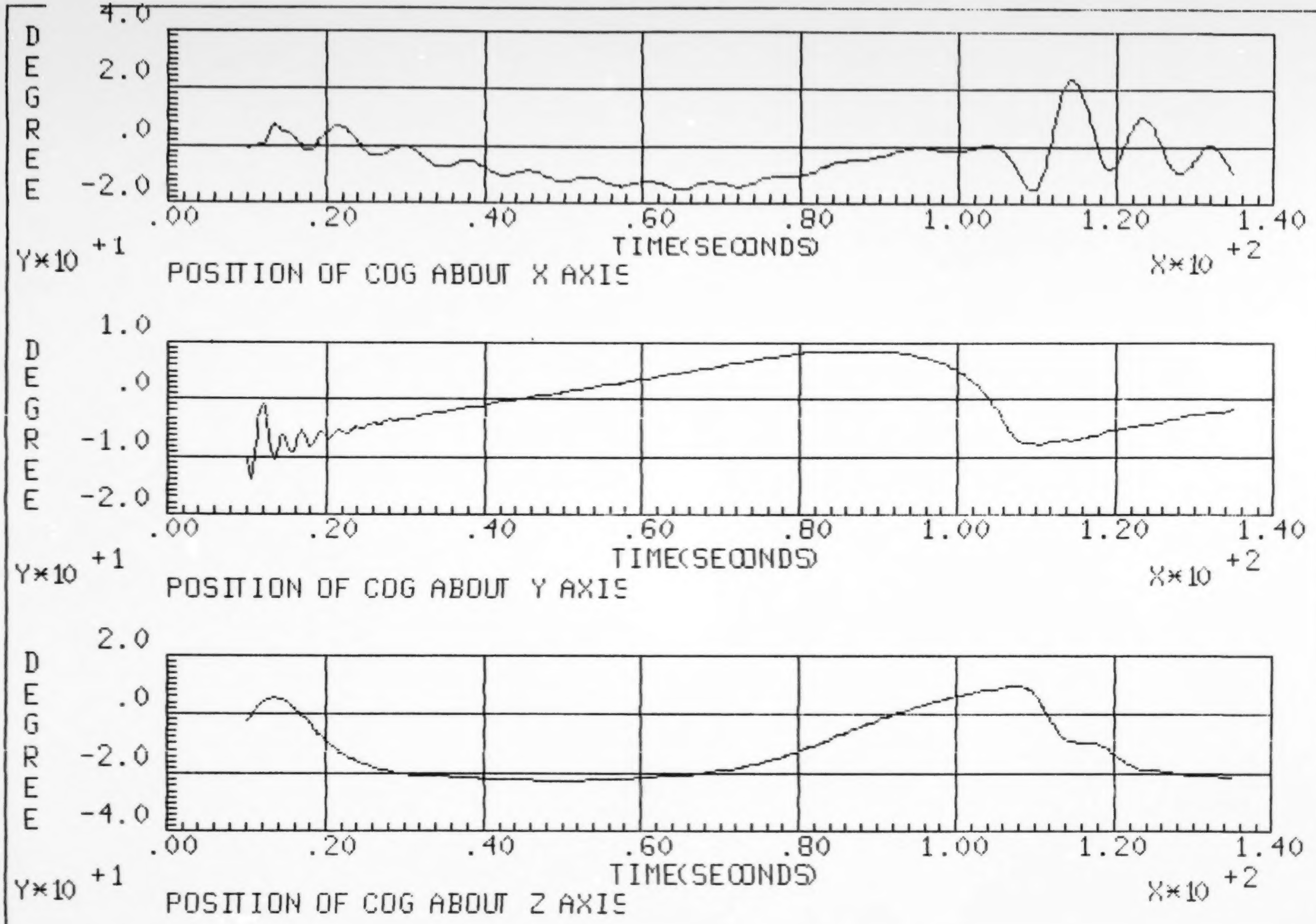

POSITION OF COG IN X DIRECTION

POSITION OF COG IN Y DIRECTION

POSITION OF COG IN Z DIRECTION

ROUND BILGE VESSEL CASE NO. ELS5

Te=100 s, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



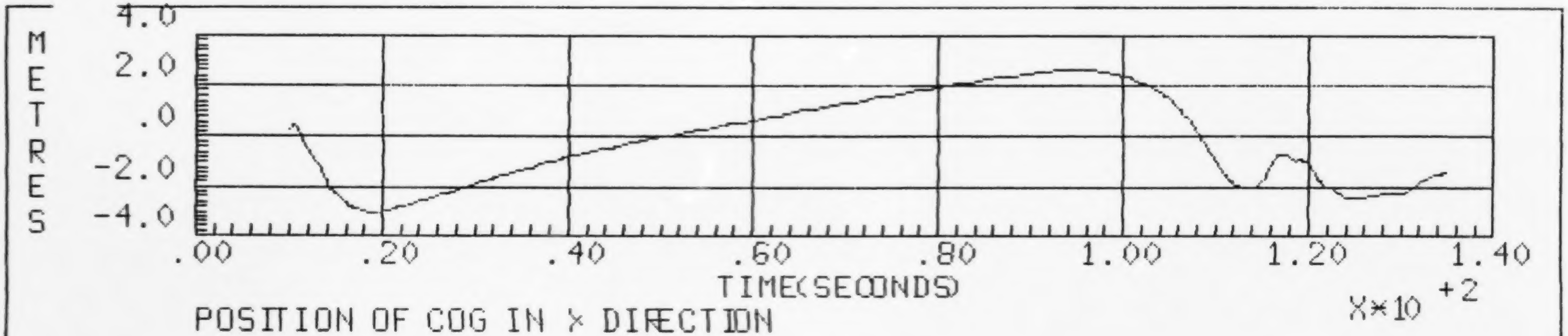
POSITION OF COG ABOUT X AXIS

POSITION OF COG ABOUT Y AXIS

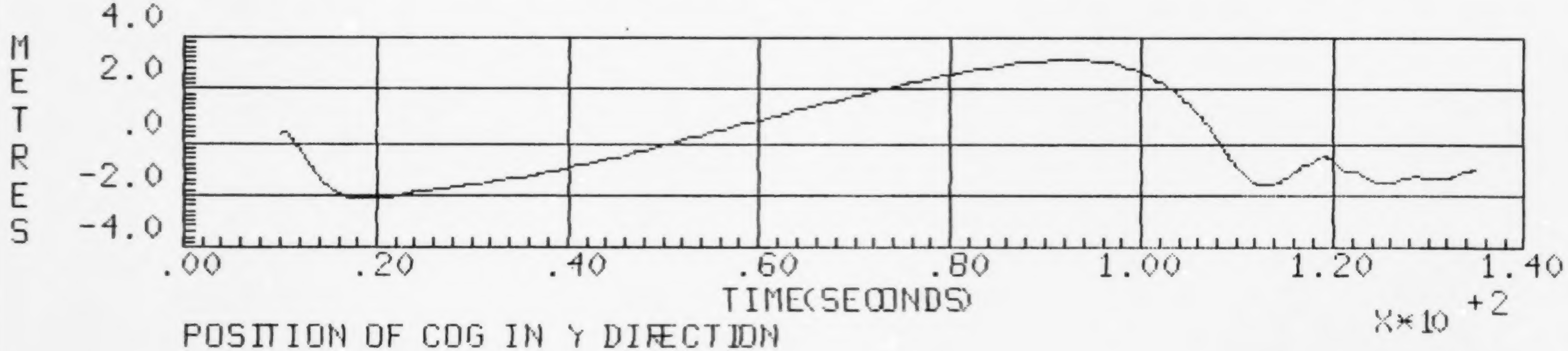
POSITION OF COG ABOUT Z AXIS

ROUND BILGE VESSEL CASE NO. ELS5

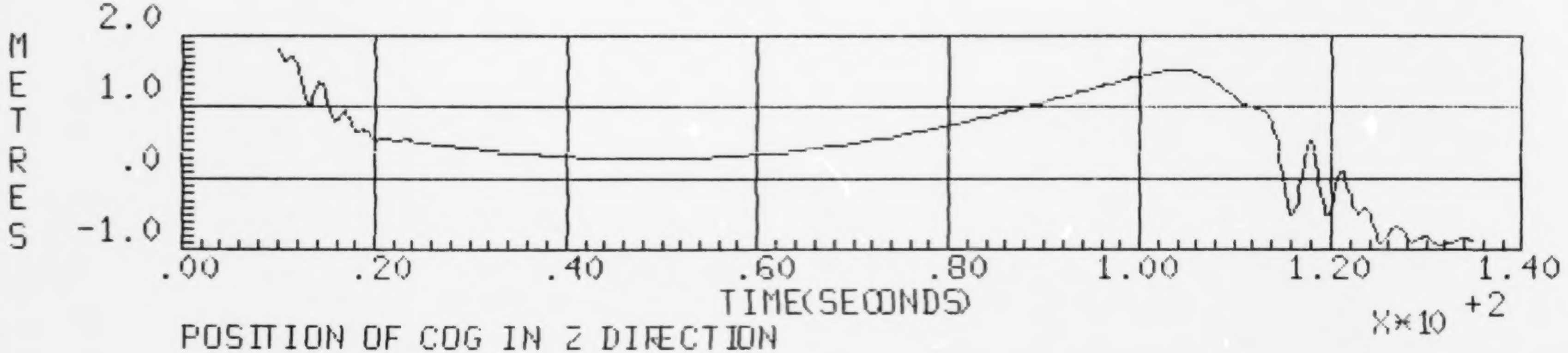
Te=100 s, Dir.=30 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



POSITION OF COG IN X DIRECTION



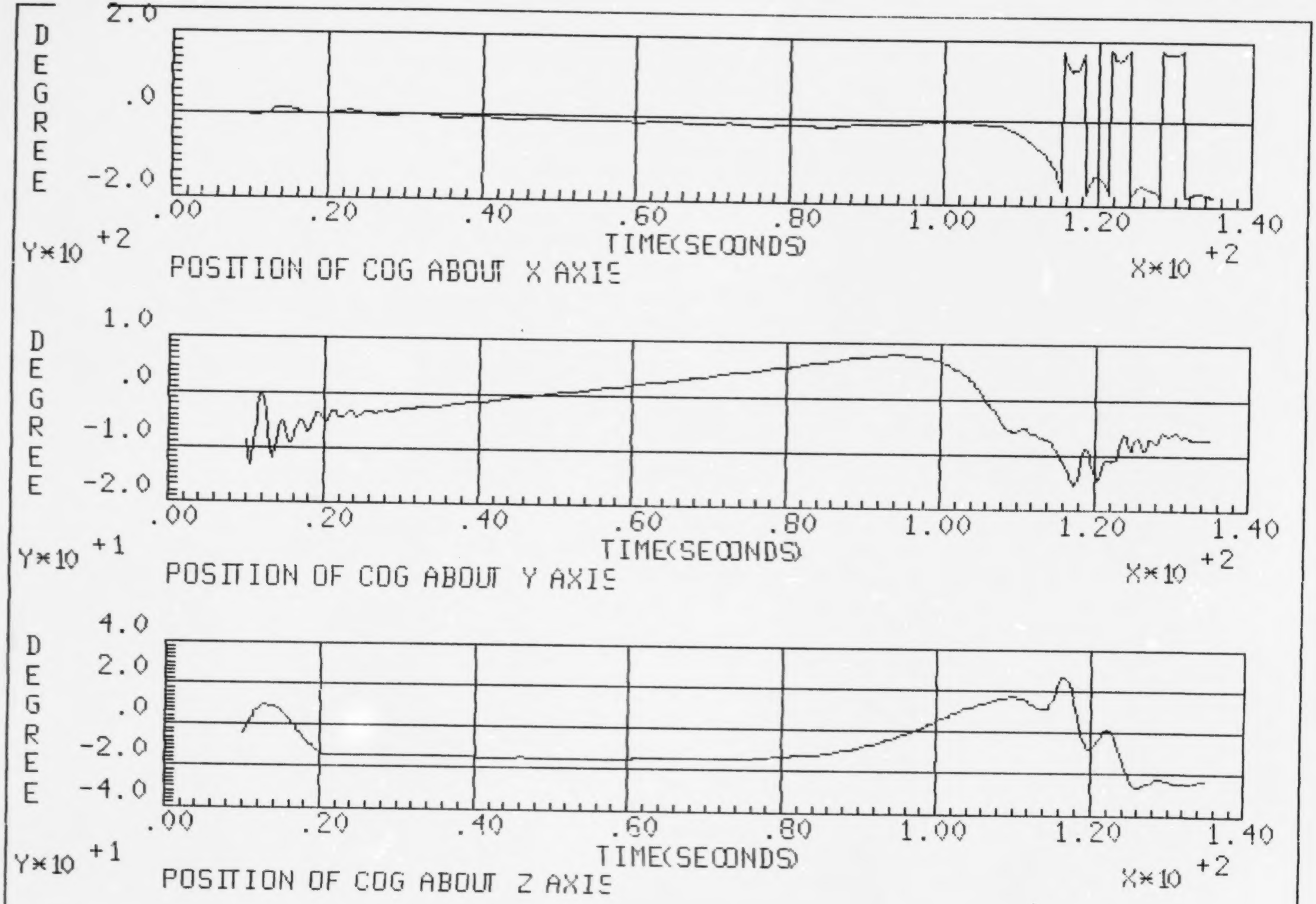
POSITION OF COG IN Y DIRECTION



POSITION OF COG IN Z DIRECTION

ROUND BILGE VESSEL CASE NO. ELS6

Te=100 s, Dir.=50 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2



ROUND BILGE VESSEL CASE NO. ELS6

Te=100 s, Dir.=50 °, H=3.24 m, Diff.=0, D_{44}=0.4 % CRIT., Servo 2